COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

September 1955

Courtesy of Brown-Boveri & Co.

Excavating for a gas-turbine power station at Jeddah, Saudi Arabia

COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

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September 1955

Feature Articles

Editorials

Departments

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BPA

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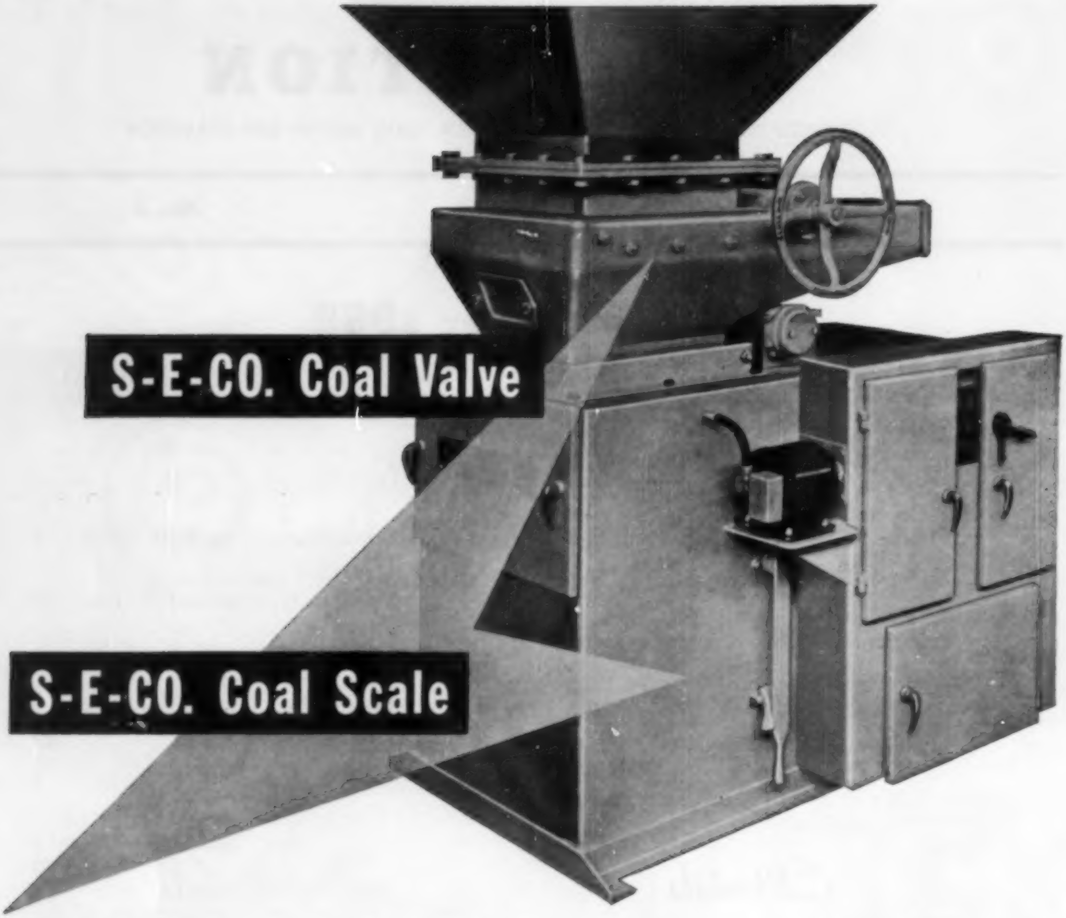
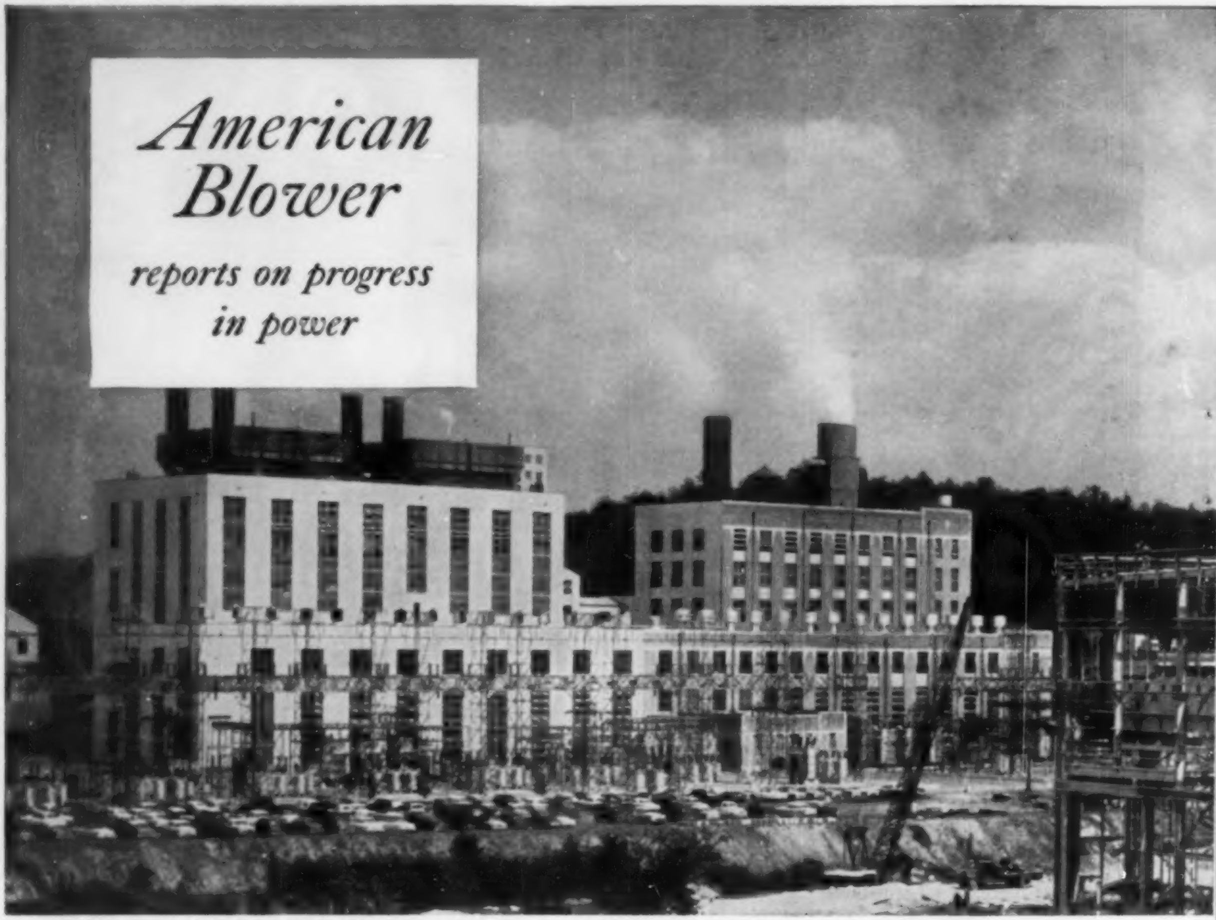
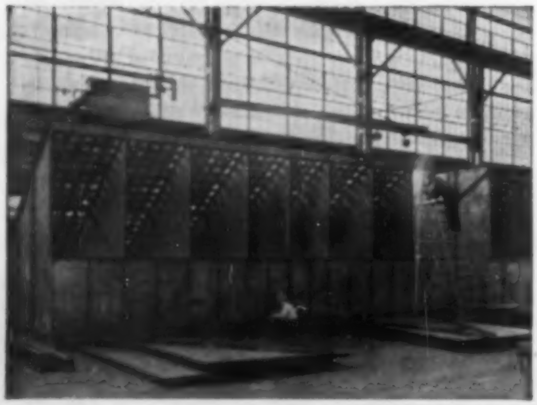

S-E-CO. Coal Valve
S-E-CO. Coal Scale



Gorgas Steam Plant No. 2, with 428,000 hp capacity, is on the Warrior River. Plant No. 3 (right), under construction, will house

Alabama Power ups capacity to



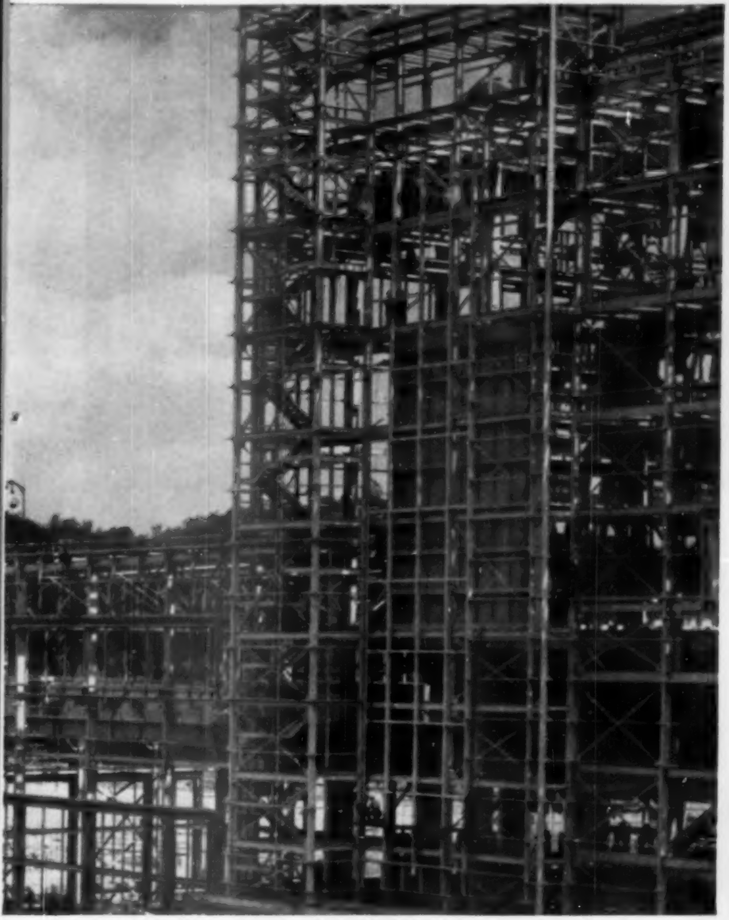
While Gorgas Steam Plant No. 3 is being built, American Blower constructs two Fly Ash Precipitators which will be installed in this new plant. Ratings: 340,000 cfm @ 285° F.

As in the past, Alabama Power installs American Blower equipment—in building for the future

With six steam and six hydroelectric plants, the Alabama Power Company forges ahead for the future of Alabama. In 1955 alone, it will invest more than $35,000,000 in bigger and better electrical facilities, including an eighth generating unit at their Gorgas Steam Plants. When completed this year, Alabama Power's capacity will be over 2,000,000 hp!

American Blower Forced and Induced Draft Fans, Fly Ash Precipitators and Gýrol Fluid Drives are being installed in this new plant.

From coast to coast you'll find American Blower playing an important role in the expansion and modernization of many other progressive, investor-



No. 8 generating unit, rated at 208,000 hp.

over 2,000,000 hp!

owned utilities. Plant operators have come to rely on American Blower Heavy-Duty Steam Coils and Fly Ash Precipitators, as well as our Mechanical Draft Fans, Dust Collectors, and Gýrol Fluid Drives for boiler-feed pump and fan control.

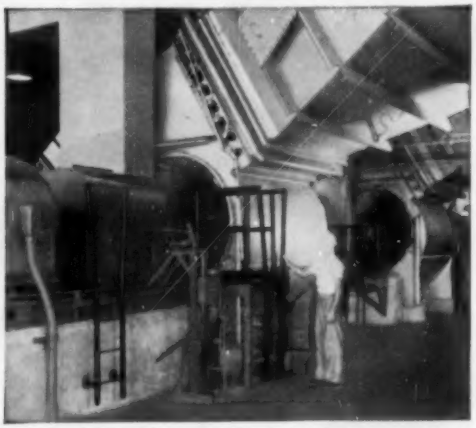
Give us a call to discuss *your* program. An experienced representative will gladly go over your requirements, and recommend equipment of the highest efficiency and economy. Contact your American Blower or Canadian Sirocco Branch Office.

AMERICAN BLOWER CORPORATION, DETROIT 32, MICHIGAN
CANADIAN SIROCCO COMPANY, LTD., WINDSOR, ONTARIO

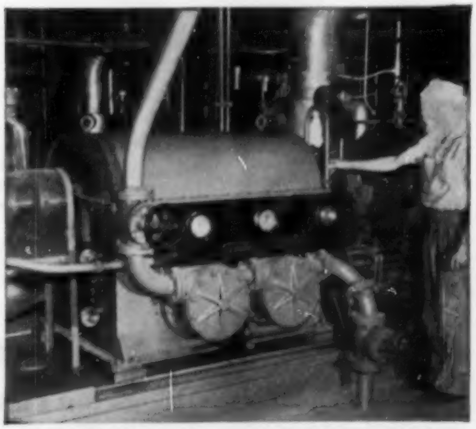
Division of American Radiator & Standard Sanitary Corporation

AMERICAN BLOWER

Gorgas Unit No. 7 uses American Blower Fans with Gýrol Fluid Drives. Induced Draft Fans (above) are rated @ 262,500 cfm @ 270° F @ 15.25" sp @ 569 rpm.



American Blower Forced Draft Fans on Unit No. 7 at Alabama Power's Gorgas Steam Plant are capable of 180,000 cfm @ 140° F @ 11.00" sp @ 868 rpm.



In addition to fan control, American Blower Gýrol Fluid Drives — class VI, 1500 hp — are used for adjustable speed control of boiler-feed pumps.

EASY TO SERVICE...

a plus value of ASHCROFT *Maxisafe* DURAGAUGES

Turn the knob on the back of the *Maxisafe* and the plate comes off, fully exposing the entire mechanism for fast and easy inspection, recalibration or adjustment. This time and money-saving feature is highly favored wherever Ashcroft *Maxisafe* Duragauges are in service.

This Duragauge was named *Maxisafe* because it has an integrally-cast wall to separate the dial from the movement and Bourdon Tube assembly — a solid wall of safety that protects the viewer if the tube ever ruptures. Covering the back of the case is a double spring mounted safety release plate — a Teflon-coated plate tightly fitted on a rubber gasket and held in place by a knurled knob. Less than 0.5 psi pressure created by a ruptured tube forces this cover open — vents the discharge safely to the rear.

The Maxisafe is available in 4½", 6" and 8½" dial sizes. You can have a choice of standard Ashcroft Duragauge pressure ranges, case designs (except Types 1179 and 1279), and mountings. Get the utmost in sustained accuracy, durability, protection and convenient servicing. Specify Ashcroft *Maxisafe* Duragauges.

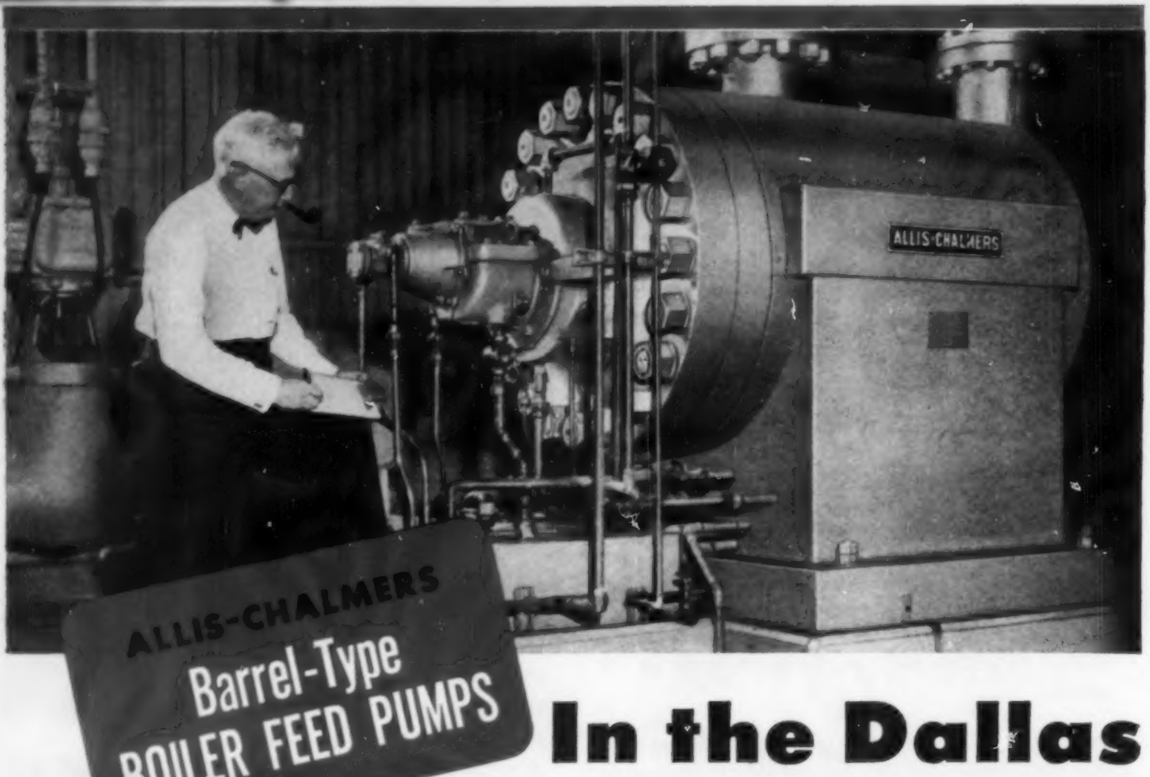
YOUR INDUSTRIAL SUPPLY DISTRIBUTOR has complete facts about Ashcroft Gauges of all types. You can depend on his knowledge and experience to help you satisfy every requirement of the application. He maintains local stocks to serve you promptly and economically.

1850 ASHCROFT U.S.A.

ASHCROFT GAUGES

A product of **MANNING, MAXWELL & MOORE, INC.** STRATFORD, CONNECTICUT

MAKERS OF 'ASHCROFT' GAUGES, 'AMERICAN' AND 'AMERICAN-MICROSEN' INDUSTRIAL INSTRUMENTS, 'HANCOCK' VALVES, 'CONSOLIDATED' SAFETY AND RELIEF VALVES, AIRCRAFT PRODUCTS. BUILDERS OF "SHAW-BOX" AND 'LOAD LIFTER' CRANES, 'BUDGIT' AND 'LOAD LIFTER' HOISTS AND OTHER LIFTING SPECIALTIES.



In the Dallas Steam Electric Station

OF THE DALLAS POWER & LIGHT COMPANY

What Are Your Pump Needs?

Whether you require boiler feed, condensate, circulating or other power plant·pumps — a careful study will reveal why more and more utilities are specifying Allis-Chalmers. Check the design features, the operating advantages, the provisions that make a pump easier to maintain, or service. You, too, will find it pays to standardize on Allis-Chalmers Power Plant Pumps.

Complete Unit from One Source

Allis-Chalmers can supply the complete pumping unit—pump, motor and control—of coordinated design and manufacture. You get one responsibility—one guarantee of satisfaction.

Get complete information on Allis-Chalmers barrel-type boiler feed pumps. Call your Allis-Chalmers District Office or write Allis-Chalmers, Milwaukee 1, Wisconsin, for Bulletin 08B7899.

Here Are Important Features that Keep Operating Costs Low:

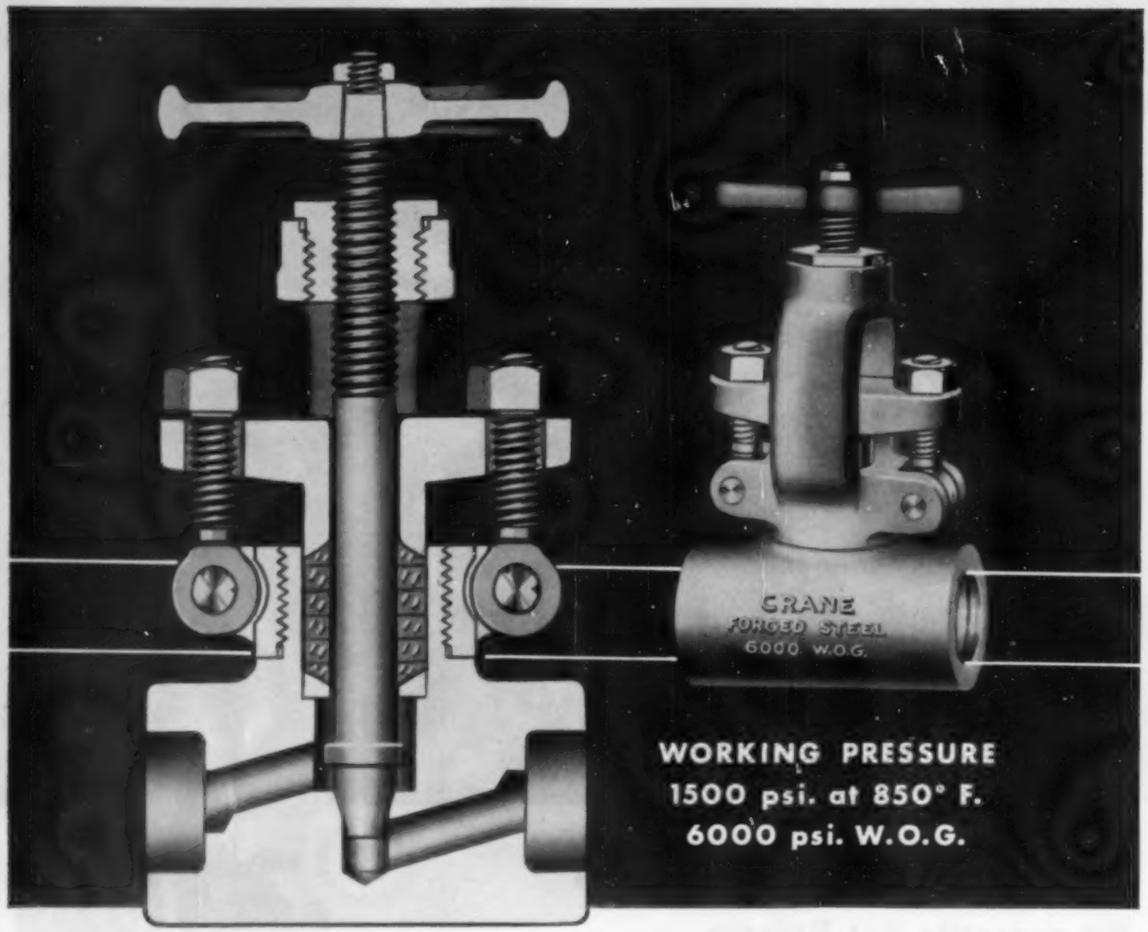
- First stage has twin, single-suction impellers to give low NPSH requirement for highest efficiency under fluctuating loads.
- Impellers, mounted back to back, balance axial forces without use of balancing drum.
- Double volute casing maintains radial balance under fluctuating load.
- Expansion joint and shaft seals are brought to outside of pump where they may be inspected often and worked on easily, if required.

A-4790

ALLIS-CHALMERS

A-C ALLIS-CHALMERS

Now Available



New Crane Forged Steel Instrument Valves

Been looking for an exceptionally rugged, compact, and low-priced stop valve for your small hydraulic or high-pressure, high-temperature instrument lines?

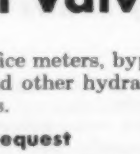
Crane has developed it—sizes ¼", ⅜" and ½", in both screwed and socket-welded patterns. It's the all-new Crane Forged Steel Instrument Valve, and you can specify it now for immediate delivery.

Will Handle Many Services

Note closely the over-all design and construction shown above of this new Crane valve. Oversized stuffing box—heavy-duty Crane Exelloy stem with integral disc—bolted gland—swinging gland eye bolts—outside screw and yoke construction—all add up to the high performance and low maintenance you want in a valve for your instrument panels, orifice meters, bypass and gauge lines, regulator leads, and other hydraulic and high pressure/temperature lines.

Literature on Request

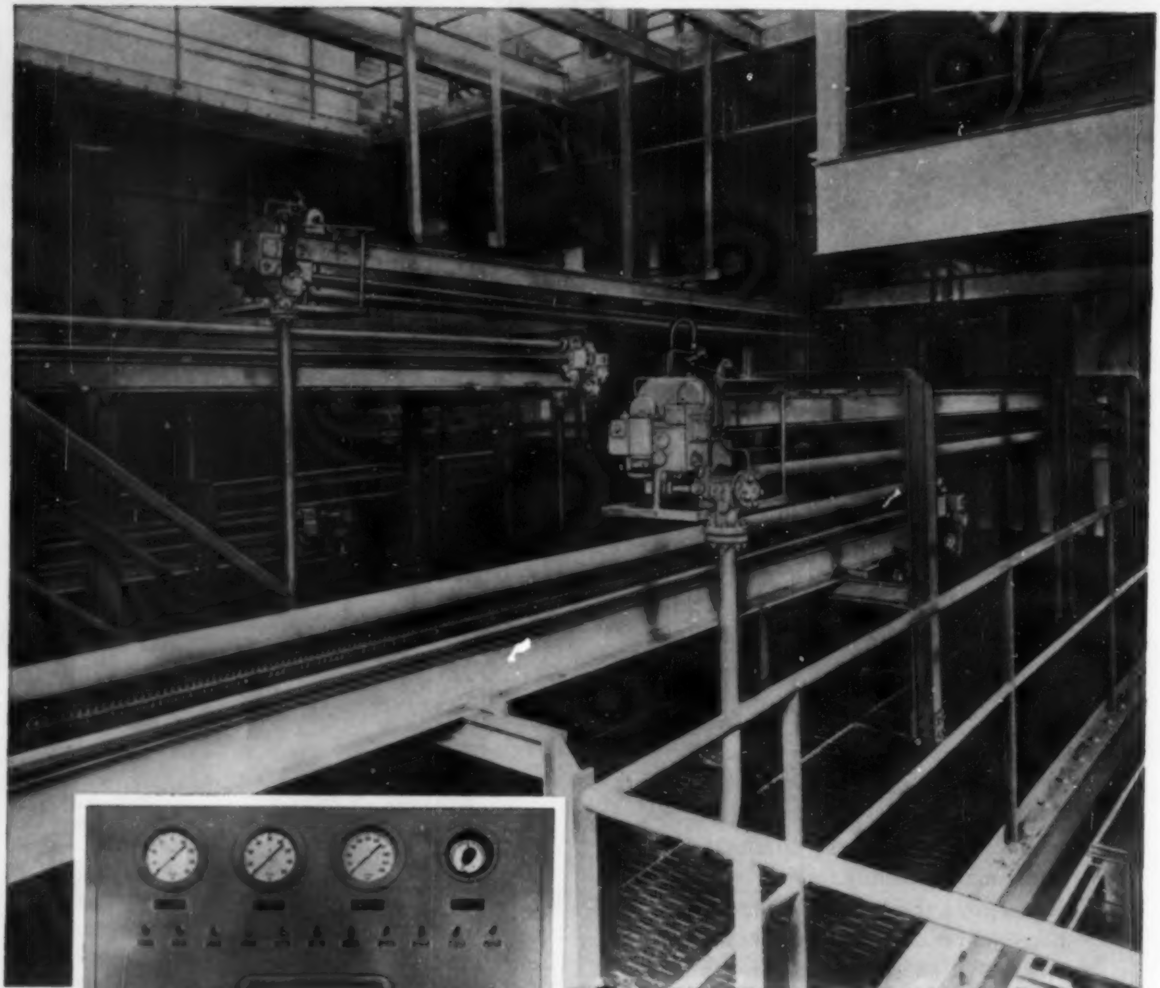
These rugged, low-cost stop valves are built for 1500 psi. at 850° and 6000 psi. W.O.G. Construction and materials are job-engineered by Crane experts—backed by a century of quality manufacturing, matchless experience. You'll want complete information on the all-new Crane Forged Steel Instrument Valves. Contact your Crane Representative, or write to Crane Co., General offices, Chicago 5, Illinois—Branches and Wholesalers serving all industrial areas.



CRANE CO.

VALVES • FITTINGS • PIPE
KITCHENS • PLUMBING • HEATING

CRANE'S FIRST CENTURY...1855-1955



and Upside Down

Vulcan Soot Blowers clean efficiently at ST. CLAIR POWER PLANT Detroit Edison Company

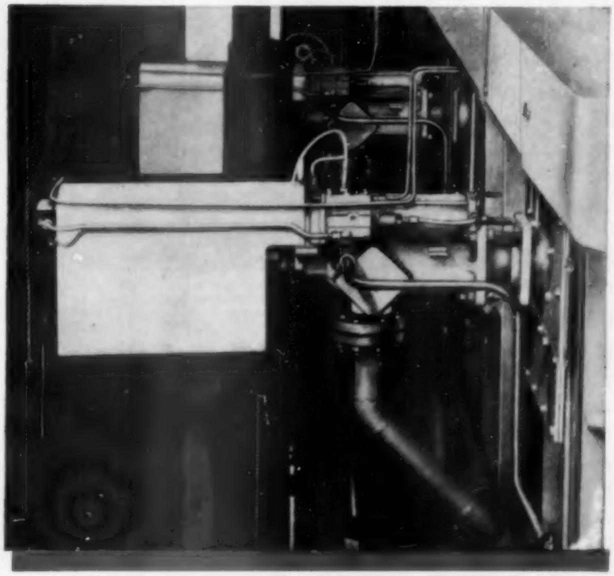
TAKE A LOOK at the big picture. Have you ever seen soot blowers nested so neatly together in such a small space? Have you ever before seen soot blowers operating upside down? These are Vulcan Long Retracts.

Vulcan Long Retracts save space outside the boiler, and require little space *inside* the boiler. The small-diameter lance fits easily into a pass with only *four inches clearance.*

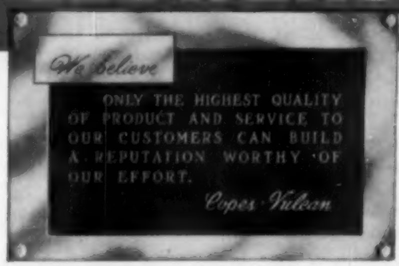
Dual-motor drive on these Vulcans—one motor for traversing, one for rotating the lance—assures optimum cleaning. The high power cleaning jet takes a different path each time it operates . . . therefore, cleans *all* surfaces *uniformly.*

These Vulcans are part of a complete Vulcan Automatic-Sequential Soot Blowing System at St. Clair Power Plant. All blowing is automatic. Just push a button, and the blowers operate in sequence, each for the proper length of time for thorough cleaning. This automatic system saves time, saves labor . . . and it is *simple.*

If you need manual, semi-automatic, or completely automatic soot blowing for large, medium-size, or small boilers, write for complete facts.



VULCAN
Automatic
SOOT
BLOWERS



COPES-VULCAN DIVISION
CONTINENTAL FOUNDRY & MACHINE COMPANY
ERIE 4, PENNSYLVANIA

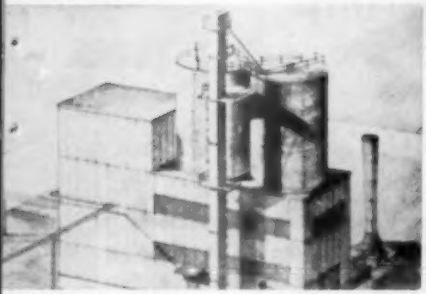
Soot Blowing, Boiler Feed Water Regulation, Combustion Control, Complete Boiler Control from one reliable source . . .

DuPont's Barksdale, Wis. Works saves $7,000 a year with new automatic combustion controls.

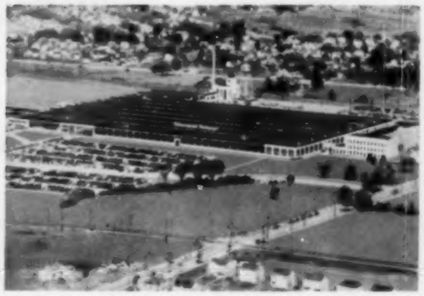
Hudson Motors' modernized plant in Detroit saves $480,000 a year—cuts powerhouse labor 27%.

Syracuse University, N. Y., built a completely new unit for maximum economy and dependability.

For efficiency... for economy...



At Stauffer Chemical's new plant in Louisville, Ky., total cost of steam is only 60¢ per 1,000 lbs.



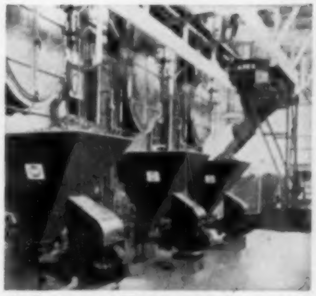
Coal costs 29.6% less than next cheapest fuel at Addressograph-Multigraph's plant in Cleveland.

Motor Products Corp. of Detroit, Michigan, saves $54,000 a year with modernized installation.

more and more firms

Burning coal the modern way saves Pennsylvania Railroad's, Ft. Wayne, Ind. terminal $33,000 a year.

Efficient new equipment reduces labor and improves performance records for Liggett & Myers at Richmond, Va.



At Lakewood Hospital, in Lakewood, Ohio, new automatic coal burning facilities cut fuel costs 22%.

Upjohn's new Kalamazoo, Mich. plant is clean and efficient with no dust or smoke nuisances.

are burning coal the modern way

Chicago's ultra-modern Prudential Building has fully automatic coal handling and burning system.

Goodyear saves $3,000 a day with new coal-burning installation at Akron, Ohio.

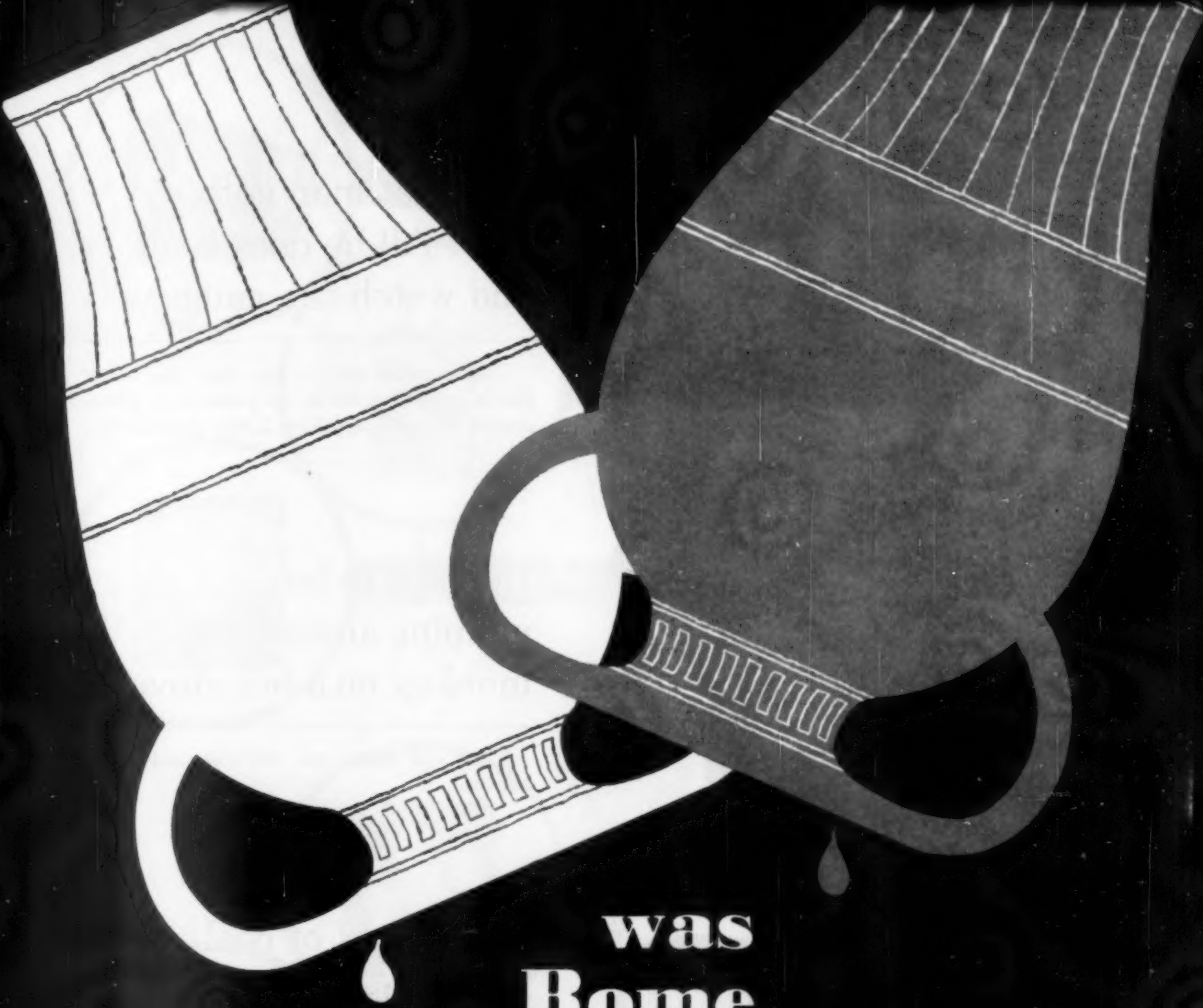
facts you should know about coal

- In most industrial areas, bituminous coal is the lowest-cost fuel available.
- Up-to-date coal burning equipment can give you 10% to 40% more steam per dollar.
- Automatic coal and ash handling systems can cut your labor cost to a minimum.
- Coal is the safest fuel to store and use. No dust or smoke problems when coal is burned with modern equipment.
- Between America's vast coal reserves and mechanized coal production methods, you can count on coal being plentiful and its price remaining stable.

For further information or additional case histories showing how other plants have saved money burning coal, write to the address below.

NATIONAL COAL ASSOCIATION

Southern Building, Washington 5, D. C.



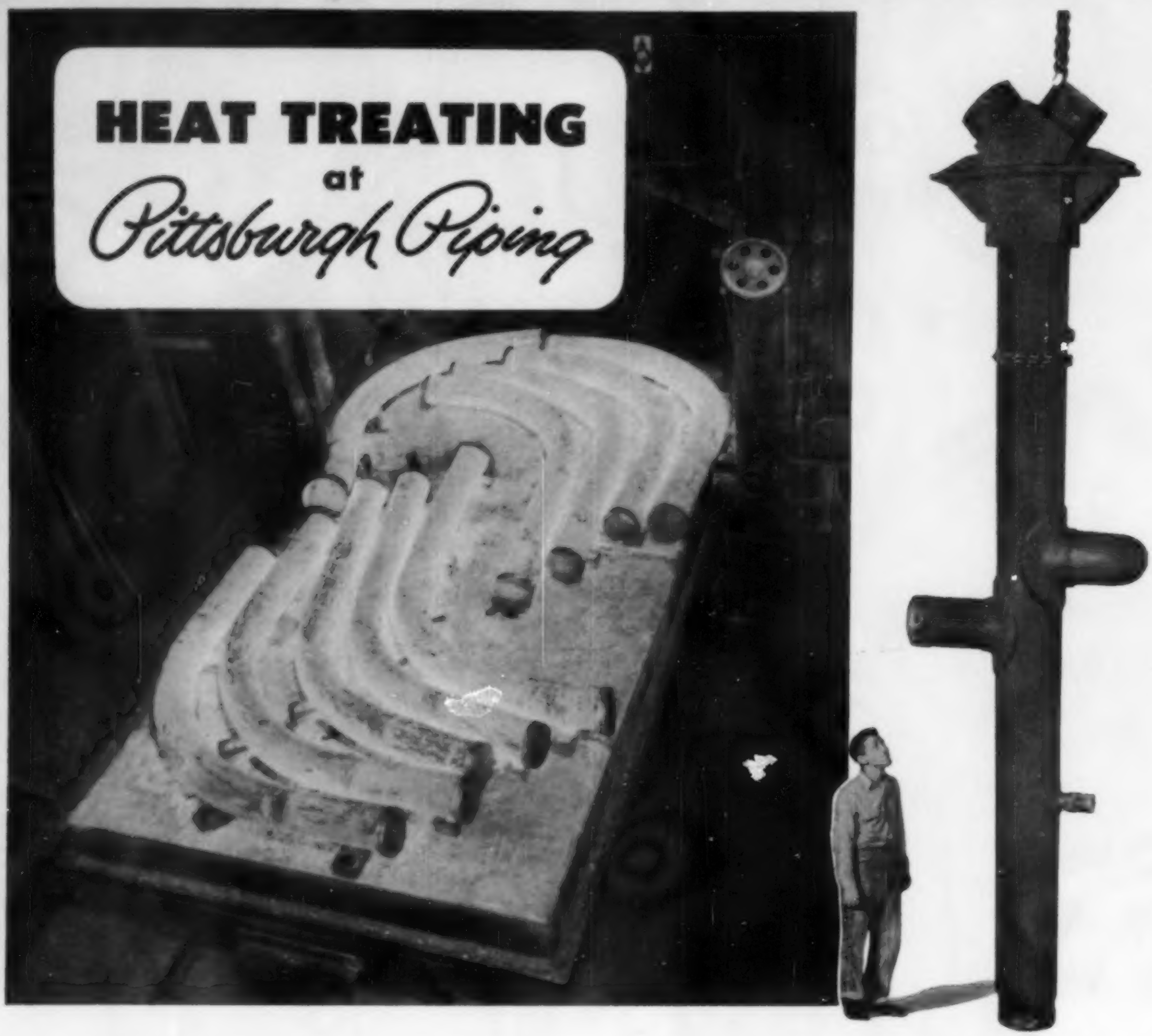
was Rome burned *up!*

Not the city, mind you, for this was before the days of the Nero caper, but the people — they wanted water. Summers were getting hotter and Romans were getting thirstier. They needed something to drink — with or without calories. Thus began the construction of Rome's famous aqueducts, one of which, scarred but faithful, served the city for almost nine hundred years!

These early aqueduct builders had more to contend with than their thirst. For one thing, they had to carry the ball alone. No skilled specialists or professional consultants stood ready to provide their answers. Today, carrying the ball alone is almost as defunct as the aqueducts themselves. Experienced consulting, design and operating engineers — each contributing his unique skills — cooperate in solving complex water and waste problems that would have amazed the early Romans. Graver believes this interrelation is not only useful but desirable and that all involved can not help but profit by it.

GRAVER

GRAVER
WATER
CONDITIONING
CO.



ASSURES PROPER TOUGHNESS, DUCTILITY, AND STRENGTH IN PIPING FOR HIGH TEMPERATURE, HIGH PRESSURE SERVICE

Heat treating and stress relieving at Pittsburgh Piping are operations which produce a uniform grain structure, assure consistent mechanical properties, and remove residual stresses throughout the length of each fabricated piping assembly.

P.P.&E.'s modern gas-fired furnace, illustrated above, is a specially-developed design. Time-temperature conditions can be electronically controlled in it for proper heat treatment of each type of metal.

This heat treating procedure is assurance that piping fabricated by P.P.&E. will have the most desirable combination of ductility, toughness, impact resistance, and strength.

PRODUCTS AND SERVICES

Carbon Steel Piping	Forged Piping Materials
Cast Iron Fittings	Headers
Cast Steel Fittings	Manifolds
Chrome-Moly Piping	Pipe Bends
Copper Piping	Stainless Steel Piping
Corrugated Piping	Van Stoning
Creased Bends	Welded Assemblies
Expansion Bends	Welded Stainless Steel Tubing
Flanges	Welding Fittings

Pittsburgh Piping
AND EQUIPMENT COMPANY
158 49th Street—Pittsburgh, Penna.
CANADA: CANADIAN PITTSBURGH PIPING, LTD.
835 BEACH ROAD—HAMILTON, ONTARIO

OFFICES IN PRINCIPAL CITIES

Atlanta	Whitehead Building
Boston	10 High Street
Chicago	Peoples Gas Building
Cleveland	Public Square Building
Houston	Heights State Bank Building
Los Angeles	1830 North Alexandria Ave.
New Orleans	P. O. Box 74
New York	Woolworth Building
Syracuse	113 So. Salina Street
Toronto	68 Yonge Street
Washington	Wire Building

PP-13

POWELL VALVES . . . THE COMPLETE QUALITY LINE . . . POWELL VALVES

POWELL STEEL VALVES

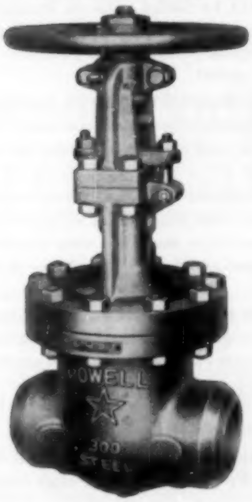


FIG. 3003 WE—Steel Gate Valve For 300 Pounds W.S.P.

FIG. 11365—Steel Pressure Seal Horizontal Lift Check Valve For 1500 Pounds W.S.P.

FIG. 11323—1500-Pound Motor Operated Steel Pressure Seal Gate Valve

FIG. 1314-A—1500-Pound Integral Bonnet Steel "Y" Valve

POWELL VALVES . . . THE COMPLETE QUALITY LINE . . . POWELL VALVES

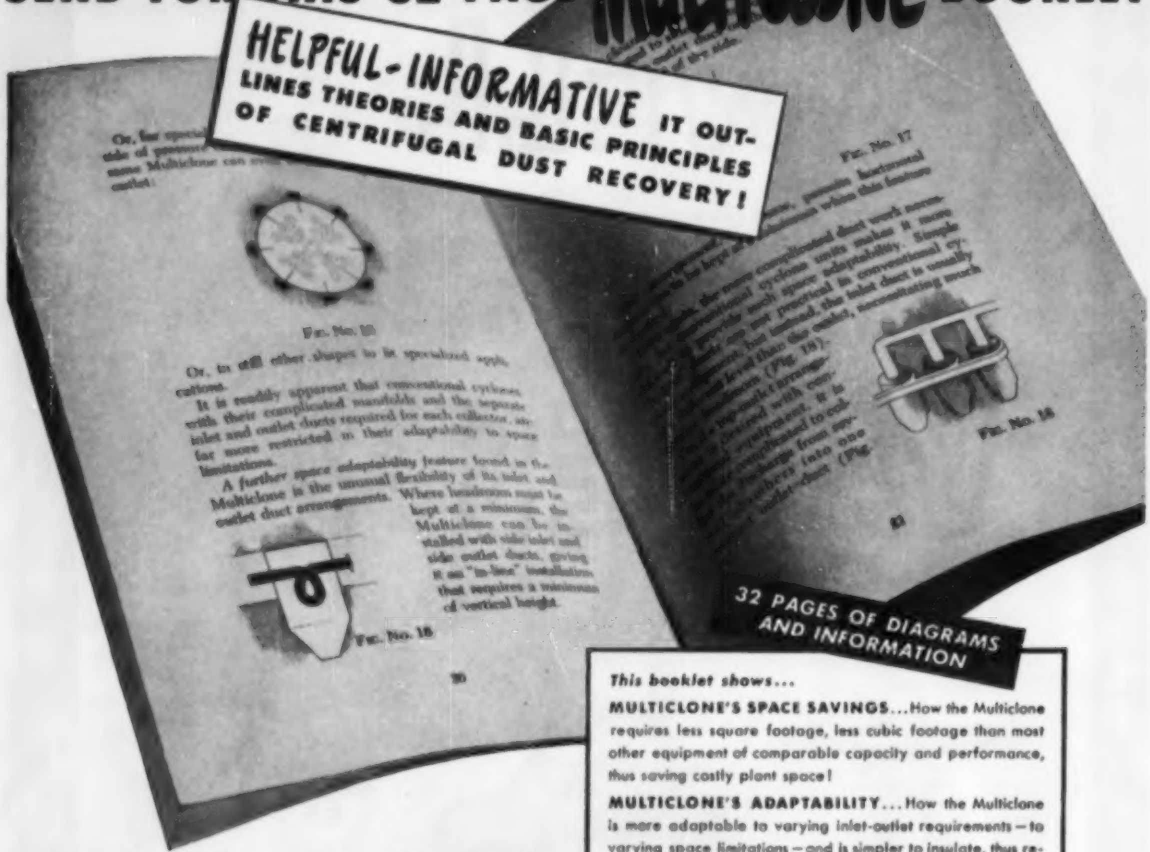
Wherever flow requires dependable control, there's the place for Powell Valves. Powell can supply just the valve you need, for Powell probably makes more kinds of valves and has solved more valve problems than any other organization in the world.

Shown above are just a few Powell Steel Valves. Investigate their many outstanding features . . . *and the complete line* of quality valves famous for dependable service.

Consult your Powell Valve distributor. If none is near you, we'll be pleased to tell you about our complete line, and help solve any flow control problem you may have. Write . . .

The Wm. Powell Company,
Cincinnati 22, Ohio . . . . . 109th year

SEND FOR THIS 32 PAGE MULTICLONE BOOKLET



NO MATTER whether you are now using mechanical dust recovery equipment or are planning the installation of such equipment at some future date, here is a booklet that is full of helpful and valuable information on centrifugal dust recovery. It not only explains the basic methods and principles involved, but also shows the important differences between small and large diameter separating tubes, shows how to simplify your duct work and reduce installation costs, and outlines many other important factors to be considered in selecting mechanical dust recovery equipment.

In addition, this informative booklet illustrates and explains how MULTICLONE'S unique *vane* design is fundamentally different... how it makes possible greater compactness, simpler installation, high recovery of the small particles as well as the medium and coarser ones, and many other facts on MULTICLONE advanced design.

A limited supply of these booklets is available for free distribution to those interested in mechanical recovery equipment and methods. Write for your copy today.

This booklet shows...

MULTICLONE'S SPACE SAVINGS... How the Multiclone requires less square footage, less cubic footage than most other equipment of comparable capacity and performance, thus saving costly plant space!

MULTICLONE'S ADAPTABILITY... How the Multiclone is more adaptable to varying inlet-outlet requirements—to varying space limitations—and is simpler to insulate, thus reducing installation costs!

MULTICLONE'S EFFICIENCY... How Multiclone's multiple *small diameter* tubes, made possible by its exclusive vane design, give higher centrifugal forces and more complete cleansing of *all* suspended particles—even small ones of 10 microns and less!

MULTICLONE'S LOW MAINTENANCE... How the Multiclone has no high speed moving parts to repair or replace, no pads or filters to clean or renew, nothing to choke gas flow or increase draft losses as suspended materials are recovered. Multiclone draft losses remain uniformly low—recovery efficiencies uniformly high—at all times!

Make sure that a copy of this helpful booklet is in your reference files by sending for your copy now!

NOW SELLING...

COTTRELL Precipitators
MULTICLONE Collectors
TURBULAIRE Spray Dryers

... in all parts of the U. S. A. and foreign countries.

WESTERN
Precipitation
CORPORATION

DESIGNERS AND MANUFACTURERS OF EQUIPMENT FOR COLLECTION OF SUSPENDED MATERIALS FROM GASES & LIQUIDS

Main Offices: 1022 WEST NINTH STREET, LOS ANGELES 15, CALIFORNIA
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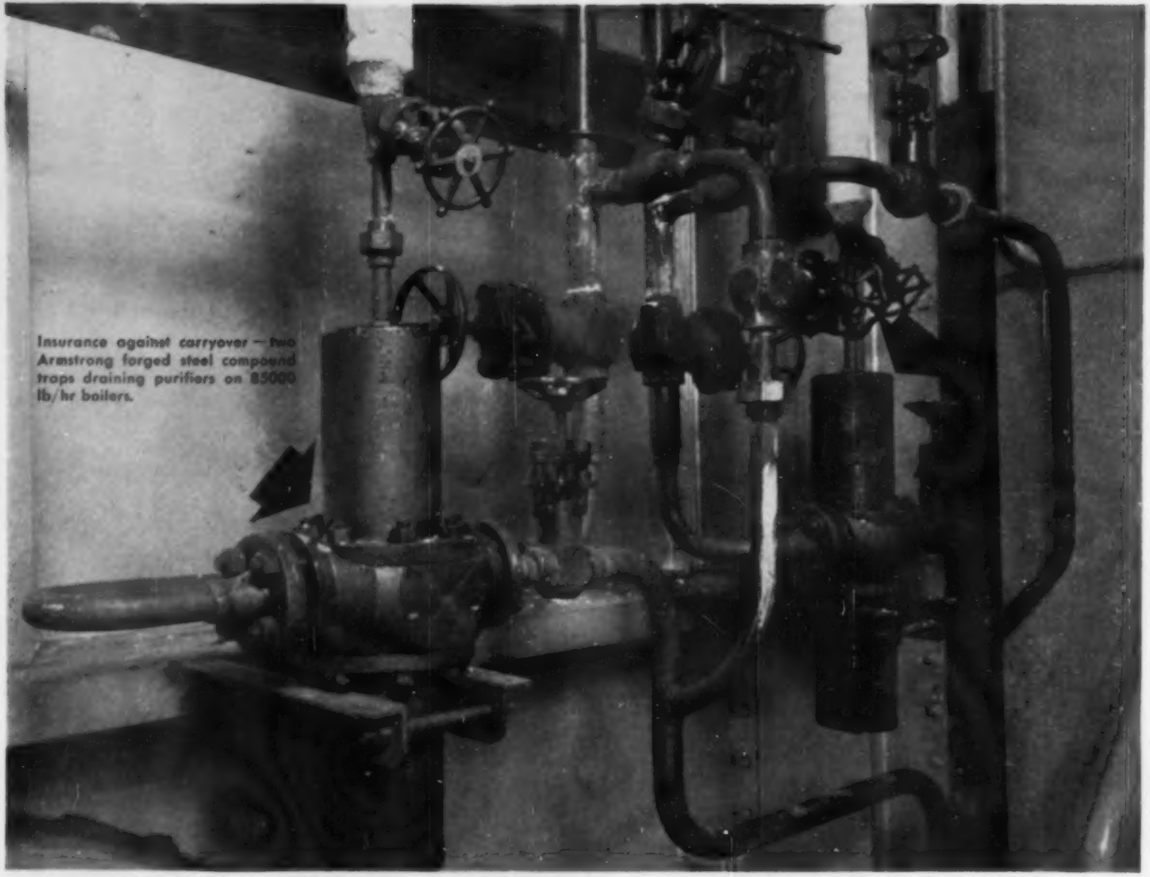
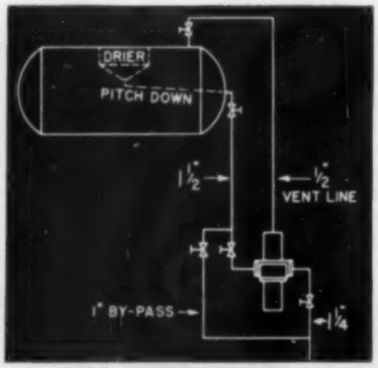
PRECIPITATION CO. OF CANADA, LTD., DOMINION SQ. BLDG., MONTREAL



Insurance against carryover — two Armstrong forged steel compound traps draining purifiers on 85000 lb/hr boilers.

How to Insure Positive Drainage of Purifiers, Large Separators and Other Big Units . . .



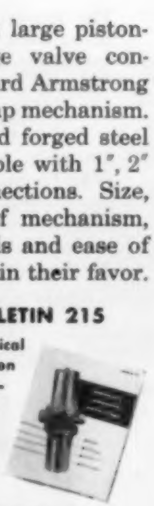
Automatic, dependable drainage of steam drier in industrial power plant with Armstrong compound steam trap.

● When you run up against a carryover or condensate load too large for ordinary traps to handle safely, Armstrong Compound Steam Traps are an ideal answer. They provide automatic, dependable drainage of loads up to 240,000 lbs/hr at 600 lbs. pressure. They have been thoroughly proved in service on purifiers, separators, driers, storage type hot water heaters, evaporators, vacuum pans and other large units.

Exceptionally high capacity in a relatively small trap is accomplished by a large piston-operated discharge valve controlled by a standard Armstrong inverted bucket trap mechanism. Cast semi-steel and forged steel models are available with 1", 2" and 3" pipe connections. Size, price, simplicity of mechanism, quality of materials and ease of installation are all in their favor.

ASK FOR BULLETIN 215

. . . contains complete physical data, prices and installation notes. Call your local Armstrong Representative or write: Armstrong Machine Works, 814 Maple St., Three Rivers, Michigan



ARMSTRONG

STEAM TRAPS FOR EVERY REQUIREMENT — APPLICATION ENGINEERED

STEAM TRAPS

Groceries . . . miracle of BRICK

Colorful cans, gleaming bottles, eye-catching cartons riding in the cart—not a single one of them would be there were it not for refractory brick. Brick that makes steel. Brick that makes glass. Brick that makes paper.

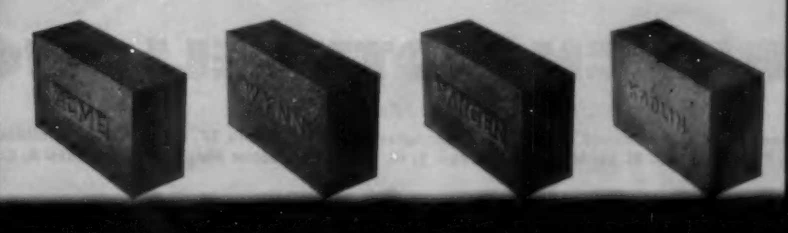
Everything that's made and everything that's moved starts with refractory brick in one form or another, lining the furnaces of industry. The requirements are almost infinite in variety.

To provide the truly complete refractory service that industry needs, General Refractories employs the world's largest refractories research laboratory; brings materials across oceans; manufactures in plants strategically located to serve industry swiftly . . . dependably . . . economically . . . everywhere.

GENERAL REFRACTORIES COMPANY, Philadelphia 2

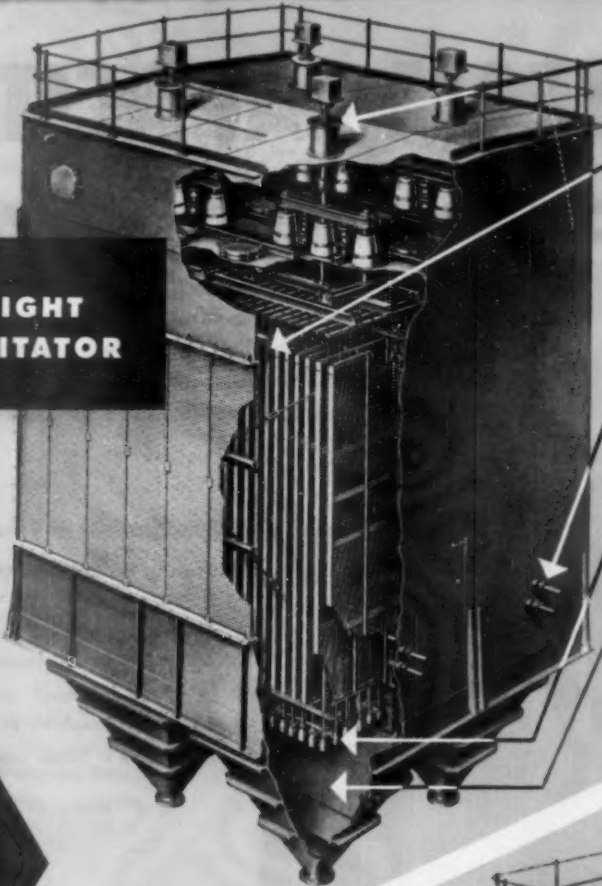
A COMPLETE REFRACTORIES SERVICE

FOR STEAM GENERATION . . . First quality fireclay brick plants strategically located across the U.S.



GENERAL REFRACTORIES

STRAIGHT PRECIPITATOR



1 Continuous, automatic discharge electrode rapping insures optimum performance.

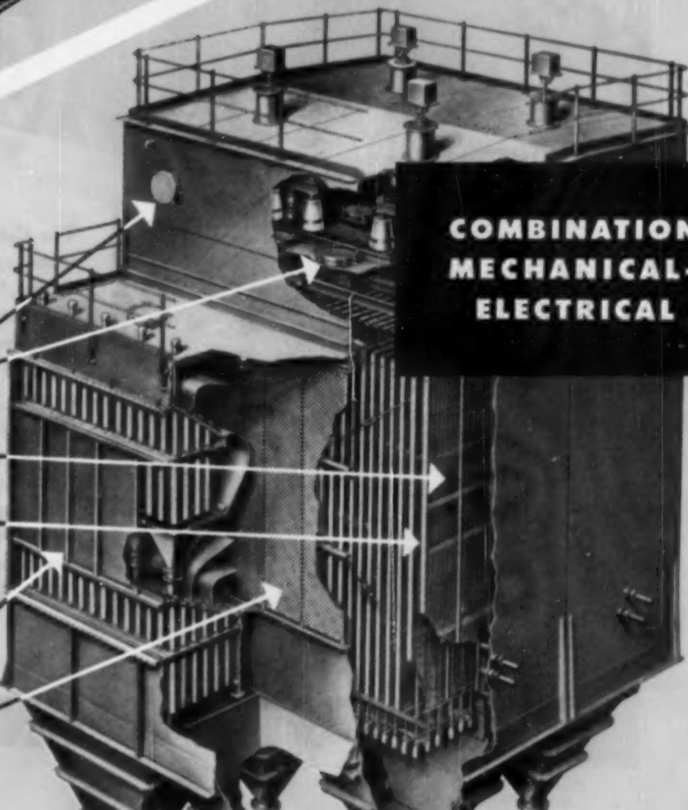
2 Hook-type collecting electrode hangers, quickly installed, insure uniform distribution of rapper vibration throughout entire collecting surface.

3 Rapping puffs prevented by automatic and continuous cleaning of collecting electrodes with Magnetic Impulse Rapper (patents pending).

4 Plumb-bob discharge electrodes are individually weighted and positioned by a steadying frame for maximum efficiency.

5 Hopper baffle prevents gas sneakage around electrodes.

COMBINATION MECHANICAL-ELECTRICAL



6 Easy access to high tension equipment.

7 No moving parts in gas stream—minimizes maintenance.

8 End and intermediate baffles provide quiescent zones and prevent re-entrainment of collected dust.

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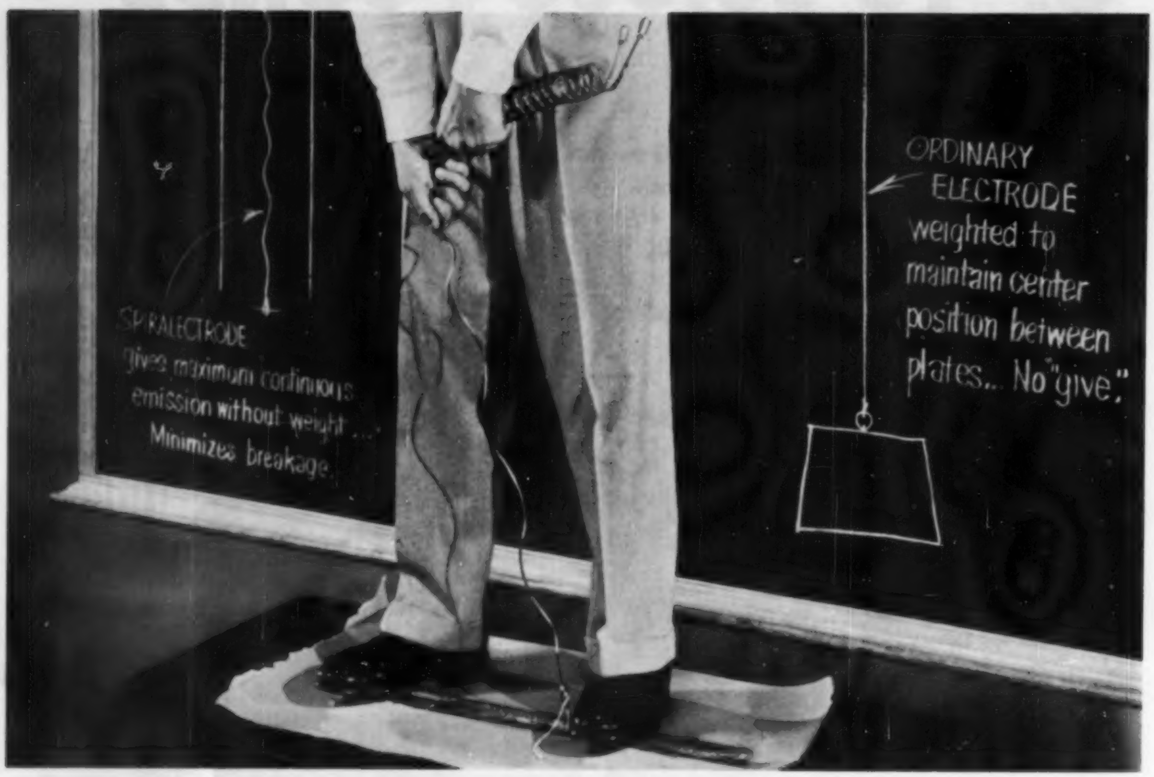

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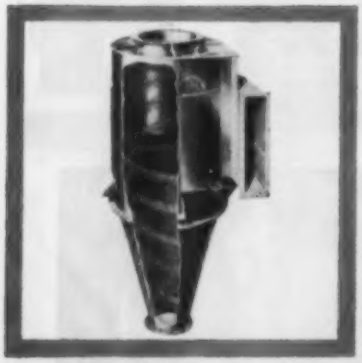

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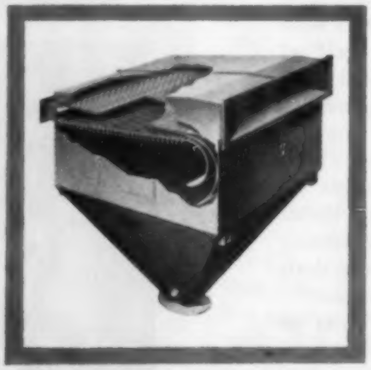

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COMBUSTION

Editorials

The Walsh-Healey Act in the Coal Fields

On August 6 the U. S. Department of Labor proposed a set of minimum wages to be adopted under the Walsh-Healey Act in the twenty-two coal producing districts. The Walsh-Healey Act, if you remember, applied to government contracts in excess of $10,000. The Secretary of Labor, James P. Mitchell, under the terms of this act has proposed from $1.40 to $2.346 an hour as a range for wage minimums in the coal fields. Included in this set of wages is one for $2.245 an hour for the mine districts embracing Pennsylvania, Maryland, West Virginia, Eastern Kentucky, Virginia, Eastern Tennessee, Ohio and Illinois, which we are told, supply four-fifths of the total U.S.A. coal production.

There are, of course, considerable definitions as to whom is covered by these minimums and as to how they are met in the case of tonnage workers. Interested parties can submit within thirty days their exceptions to the proposed decision.

The coal industry feels that the Government has put itself into a position where it must dictate rules of behavior for itself. The contention has been made that the Government has assumed by this minimum wage action the moral responsibility of seeing that its buying policies are revised. The revisions must be such, the industry feels, that the federal coal-buying agencies will be instructed to pay the producer's average cost, at least, plus a reasonable profit for each ton of coal they buy. Prior to this U. S. Department of Labor decision the Government could take the stand that it was not responsible for the coal industry's costs. Today there is room for question and in the minds of the coal industry the Government has established in effect a floor under the price of coal.

Whatever the outcome in regards to the implications of federal price-fixing the coal industry has cause for concern. It has only been of very recent date that the country's tremendous prosperity has begun to seep through to this important segment of the economy. Added costs will delay this strengthening action. The entire power generating industry, dependent as it is on the fossil fuels of which coal is by far the major representative, will follow closely the discussions and the rulings on the application of these minimums. Certainly any step which tends to establish a rigid price structure on a basic fuel can influence the thinking on the application of this basic fuel to meet the power field's needs. It is our hope that all sides of the problem will be given a proper and thorough airing.

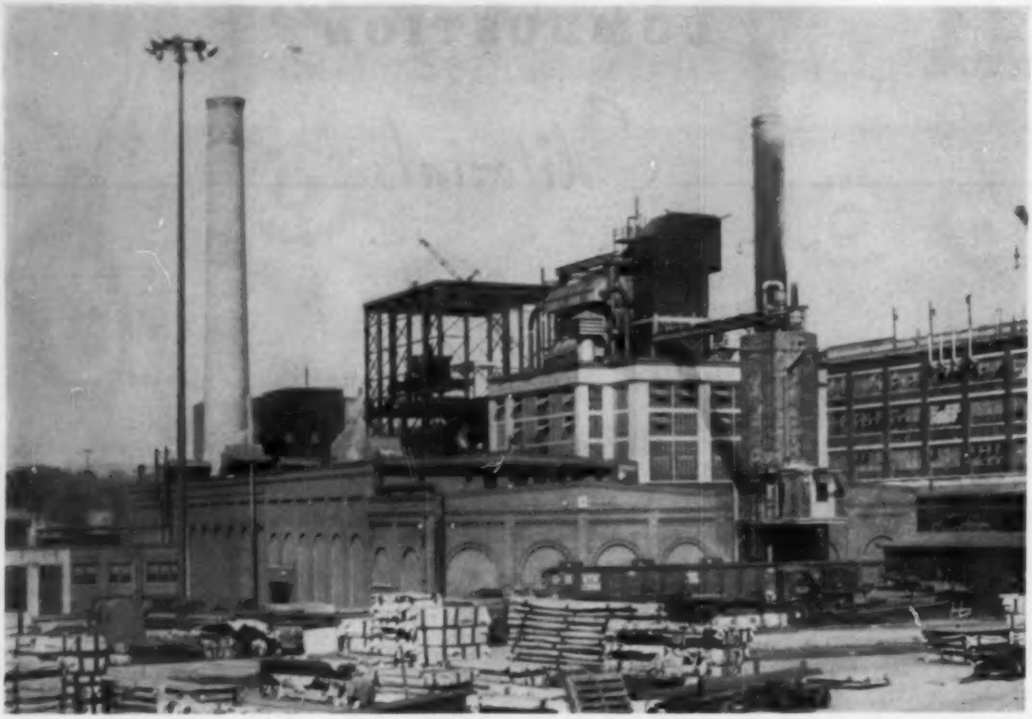
International Exchange of Information

In this issue of COMBUSTION the reader will find technical information concerning boiler circulation studies made in Germany, an analysis of cooling tower selection based on a paper originally presented before the Institution of Electrical Engineers in Great Britain, an exposition of techniques used to conserve alloy steels in Czechoslovakian boilers, a survey of nuclear power highlights of the Geneva Atomic Energy Conference (including excerpts from papers dealing with U.S.A. and U.S.S.R. power reactor programs) and an article on the scheduling of an American industrial power plant expansion.

Why does this cosmopolitan interchange of technical information take place? One answer is that sharing of information has what might be termed a catalytic effect which contributes to progress in science and engineering. But a more basic reason is that the natural phenomena which scientists and engineers study know no national boundaries. Even the most rigid security procedures cannot withhold the secrets of nature from the probing intelligence of man, a lesson that was dramatically retaught to many young scientists and engineers at Geneva. In their first opportunity in more than a decade to exchange scientific information with some degree of freedom, physicists on both sides of the Iron Curtain reported substantial agreement on independent and security-compartmented fundamental measurements of neutron cross-sections.

Since 1940 there have been many wide swings of the pendulum of public opinion toward sharing of information. Right now, in contrast to the earlier years of the Cold War, there appears to be a period of relaxation, if not removal, of barriers to free interchange. Paradoxically enough, the necessity for security that results from tense international relations may be dissolved by the realization that sharing of information may contribute to the relief of international tension. It is in this sense that the feeling of camaraderie that was engendered among the scientists and engineers attending the International Conference at Geneva may stand out as the most significant contribution of that gathering.

Free interchange of information contributes both to progress and to the efficient use of our intellectual resources. Scientists and engineers have a direct interest in the conservation of natural resources. They should have an equal interest in the promotion of international exchange of information as a means of conserving human intellectual resources.



Overall view of the G. E. Pittsfield, Mass., Works shows the powerhouse during the expansion program and pictures the concentration of existing neighboring buildings which added to the problems in scheduling the expansion steps

Pittsfield Power Plant Expansion Benefits From Good Scheduling

Setting a boiler room expansion program that involves a demolition phase for one section of the powerhouse beginning May 1 and a steaming date of the following January 1 for the new boiler looks like an impossible task. Here is the account of a successful powerhouse expansion that followed just this schedule.

By A. H. CHILDS*

General Electric Co.

THE Pittsfield, Mass., Works of the General Electric Company built its first powerhouse in 1901. By 1938 it had replaced three of the original boilers with two new units and also erected a compressor room addition to the powerhouse. 1941 saw one more boiler go in. Then in 1947 the mercury boiler was installed and the powerhouse still further enlarged. In the early 1950's general plant growth indicated still another increase in powerhouse capacity was at hand.

By 1953 the increased plant process requirements had reached the point where early action on the next phase of powerhouse expansion was in order. Further, the penalties created by any losses of production from power shortages or failures were sizable enough to warrant a certain reserve in capacity. General Electric Companys' decision was to install a 200,000-lb boiler. Once the decision to expand was made the earliest possible completion date became highly desirable.

It was early recognized that boiler delivery would be the key to the building construction and completion date. Exploratory negotiations were begun with a completion date of December 31, 1954, included as part of these negotiations. Boiler drawings were actually started in February even though the contract for the boiler was was not let until May of 1954 to Combustion Engineering, Inc. The reason for this head start on boiler drawings

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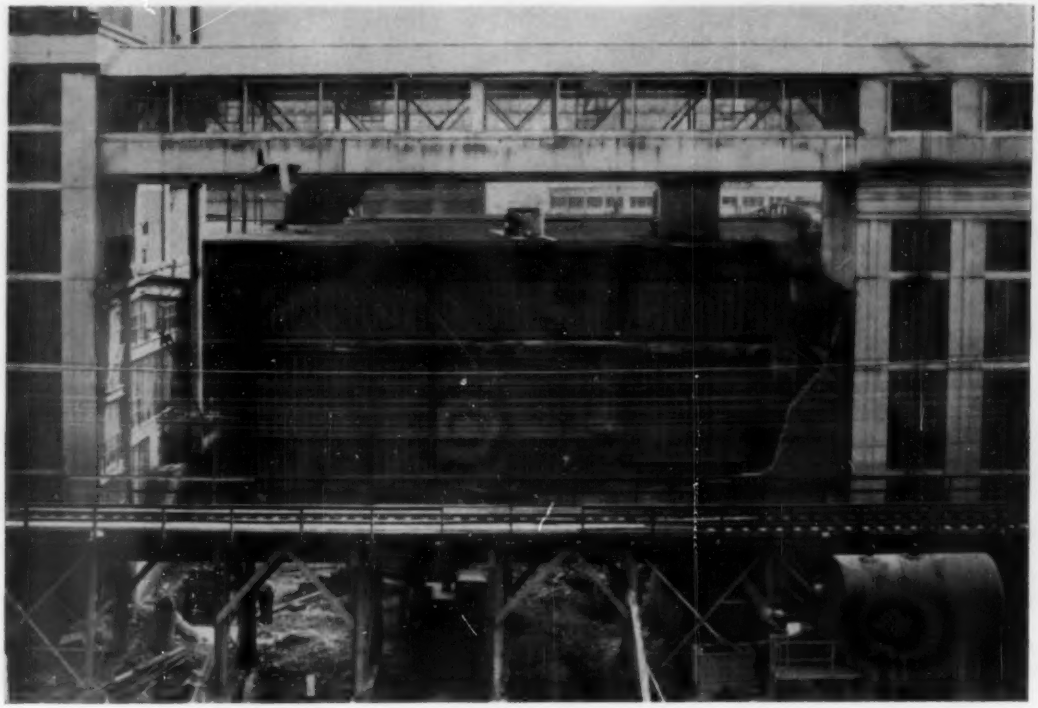


Fig. 1—Interior view of the demolition area cleared for the installation of a 200,000 lb per hr steam generator. Demolition began May 1 and was completed June 2. The pile driving contractor moved in June 3 and was underway June 8

was the fact that the boiler had to be designed to fit a specific physical area and hence could not be a standardized unit.

Early Planning

On March 17 a meeting of the interested General Electric Company personnel was held to coordinate initial planning of the engineering, demolition and construction phases of the project. The overall design and basic engineering of the powerhouse extension was assigned to the Company's Real Estate and Construction Engineering Division. This involved the preparation of all drawings, the dovetailing of new building drawings with those of the boiler manufacturer for steel work primarily and the development of all required demolition drawings. It was agreed that any necessary sheet piling vital to the demolition phase would be shown on these demolition drawings.

Further, a very important working arrangement was devised. As Figs. 1 and 3 show a railroad trestle runs along the north wall of the power station. A definite schedule for construction crews to have access to this trestle was essential. Accordingly it was planned to give these crews access during daylight hours and whenever else the Pittsfield Works' schedule would free the trestle.

Next the demolition schedule had to be set with a view toward carrying out the powerhouse expansion with as little disruption of existing plant services as possible. An outage of about two to three months on two of the turbines, No. 5 and No. 6, for example, was believed acceptable to the Pittsfield Works. This outage was a necessary one since the main turbine leads and the main relief line from No. 6 machine would interfere with and would have to be removed by the start of demolition. Reinstallation of these lines in their permanent locations would have to wait until certain of the demolition work was completed and design drawings advanced to the point where interferences between these lines and new work would be definitely past. All requirements such as these had to be anticipated in determining the demolition schedule which was finally planned to start May 1.

The Construction Engineering Section of General Electric Co. supplied a list of finished drawing dates for the various phases of the powerhouse modernization at this same meeting. For example their March and early April 1954 calender called for drawings covering these pre-demolition jobs: fuel oil piping, pumping and heating set to the mercury boiler unit, the temporary support of the 625 psi steam line, relocation of the mercury reclaimer and drain tank, temporary support of the condenser boiler relief line and the 6-in. blowdown and oil pump vents, the relocation of the 6-in. feedwater header.

In addition at this early March 17 meeting the advantages of redesigning the main power condensation system was discussed. It was felt that the existing condensate system would be inadequate for the increased power plant load. Any major cutovers involving the new condensate systems could be handled best during the summer months when loads were at their minimum. As a result design steps had to be completed to meet this summertime cutover and the necessary preliminaries were set in motion at this time.

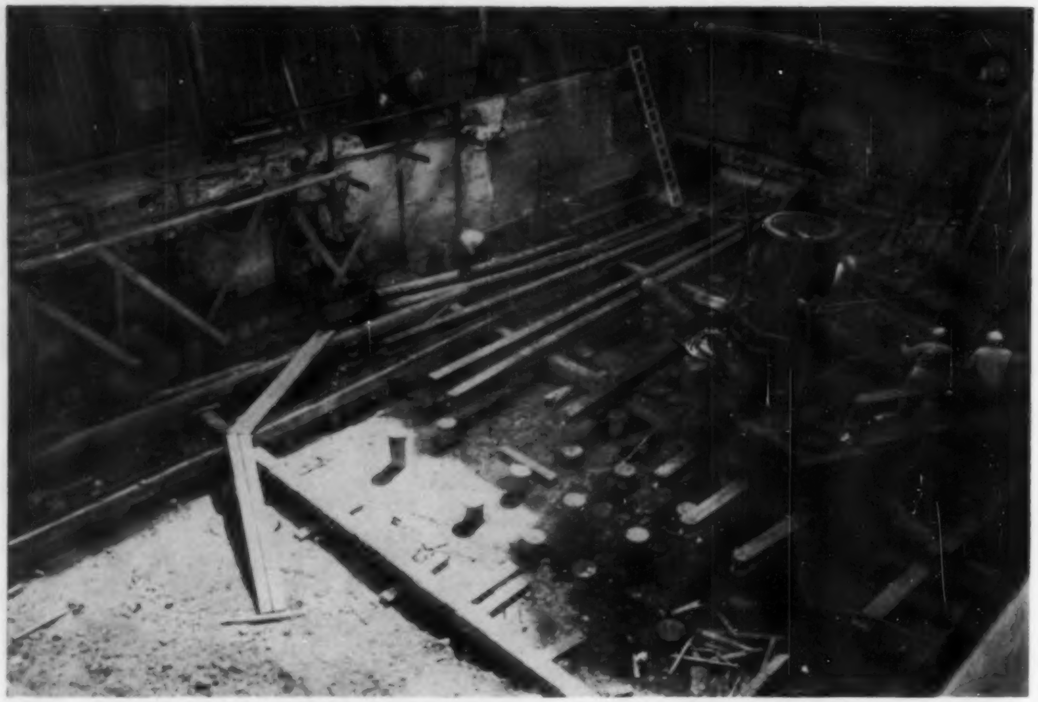


Fig. 2—The actual cross-section area made available for the plant expansion measured 50 ft by 80 ft so expansion proceeded a step at a time. This shows the pile driving phase. 146 piles were driven to an average depth of 30 ft.

Setting Schedules

With the preliminary planning and discussion over with, the boiler schedule to meet a steaming date of January 1, 1955, was laid out. The high spots involved:

Demolition started—May 1
Boiler steel shipped—July 16
Boiler steel received—July 20
Boiler steel erected—July 21–August 15
Boiler drum shipped—August 10
Boiler drum delivered—August 16
Boiler drum in place—August 20
Tubing, headers, superheaters—As soon after August 20 as practicable
Boiler hydrostatic test—November 1 (later moved up to October 20)
Boiler on the line—January 1 (later moved up to December 15)

Against this rather rigid schedule the contractor, United Engineers and Constructors, Inc., who were awarded the contract for *overall* construction, had to plan for all operations to take place in an area roughly 50 ft by 80 ft, Fig. 2. This space limitation made it necessary to carry out the major phases of the powerhouse expansion one at a time. In addition to the overall construction United Engineers undertook the responsibilities for purchasing, expediting, field work, subcontractors' services and the installation's approval.

Despite these formidable restrictions the contractor allowed time for competitive bidding although it was agreed that a lump sum award be made on major equipment so that a 40-hour week could be set up and control could be established over labor costs.

It was decided early to hold weekly meetings of the architect, operating personnel, the contractor and the General Electric Company. In this way progress could be closely gaged. Equipment delivery delays were investigated. When necessary, visits to the supplier's factories were made to approve a troublesome design point or permit fabrication changes.

Demolition Details

The boiler steel shipment date of July 16 permitted roughly two months for the demolition and pile driving phases of the powerplant expansion program.

The demolition operation as mentioned above involved the rerouting of interconnecting steam and water lines running through the old building to either of the later additions. As a result the equipment in this rerouting had to be delivered in a matter of weeks so that the contractors took special steps to aid in expediting these items.

The clearing of the site went ahead in straightforward fashion, Figs. 1 and 3. The physical task itself, however, was a considerable one. The original slab of roughly 50 ft by 80 ft ran as much as 4 ft thick in places. As a result the demolition contractor went on two shifts and eventually three-shift operation. This work schedule required advanced planning whenever use of the coal unloading trestle was needed.

Driving of sheet piling started right on the heels of the site-clearing, Fig. 2, as the area opened up. In fact by May 20 the south wall sheet piling was all in place.

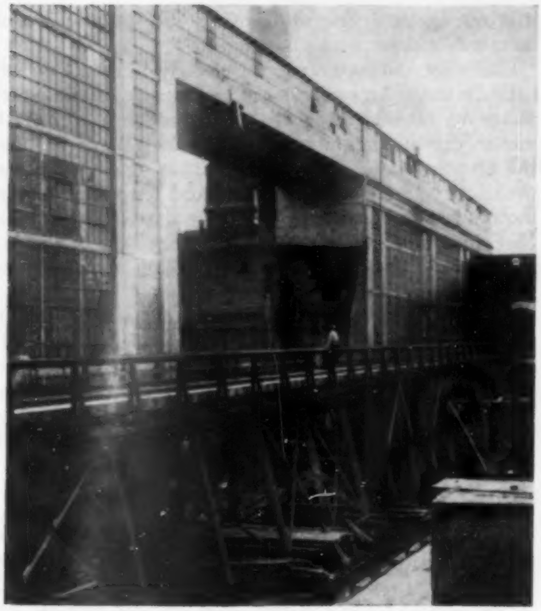


Fig. 3—Railroad trestle runs along the north wall of the power station and its continued availability was essential to general plant operation so its use had to be scheduled

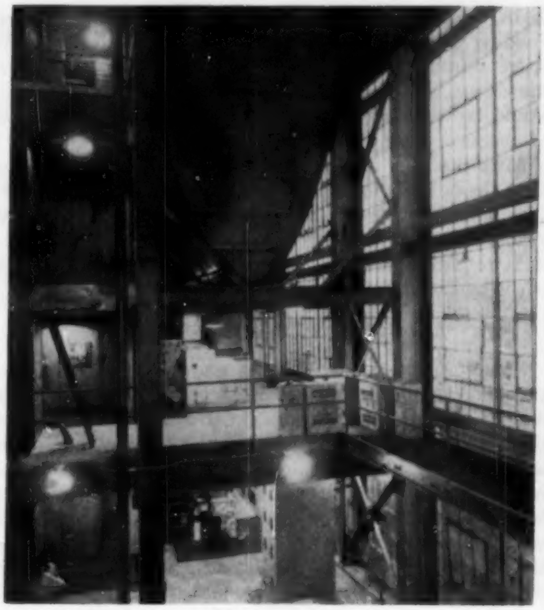


Fig. 4—Completed boiler installation, as it looks today shows in the above photo and the shot was taken within a few days of one year after Fig. 2

It was agreed that all phases—site clearing, sheet piling—were to be pushed around the clock so that the area would be ready for the piling contractor to move in May 27. Actually the demolition was completed June 2; the contractor moved out June 3; and the pile driving contractor began unloading and erecting equipment June 3.

Pile Driving Phase

The decision was made that grading and placing of reinforcing steel would closely follow pile driving. Accordingly a contract was awarded June 3 to the Bethlehem Steel Company for reinforcing steel with a firm promise of delivery June 15. At about the same time the foundation concrete vendor was chosen (May 28) and directed to purchase 4000 bags of cement immediately.

In the meantime the relocation of the 625 psi steam header from the mercury boiler was moving right along. The high temperature insulation was completed June 9 and the mercury boiler was started up 8:00 a.m. of that day after having been shut down to permit piping changes.

The pile-driving contractor had begun driving piles June 8. By the time this operation was completed 146 piles had been driven to an average depth of 30 feet and within a range of from 24 feet to 40 feet deep. All of this was finished by June 29 and the contractor moved off the job site July 2.

The fabrication of the foundation mat forms, scheduled to closely follow the pile driving, was well underway by June 18 and the drainage installation was started at about this time. Everything was ready for grading by June 28 and the grading completed July 7. The reinforcing steel was ready to go in June 30 with the entire entire foundation mat and wall finished by July 15.

During this period all the shop details comprising some thirty-six sheets for the structural steel work were being checked and approved. The expeditor's report received July 8 showed about 40 per cent of the structural steel was then in fabrication. The bunker steel had been delivered July 6; the first shipment of the building framing arrived July 9; and the boiler supporting steel had left Chattanooga, Tenn., July 8.

The piping drawings were mailed to the bidders July 8 with bids due July 22. Before the bids could be returned a special meeting was called for the various GE departments concerned and the contractor. The purpose of this meeting was to review the piping drawings and the proposed work they represented to be done during the August shutdown of the plant. Out of it was to come a schedule for the piping contractor to outline the week-end programs and the critical jobs. The primary concern was any required shutdowns for the main lines, whether or not they could be planned for weekend action, just which shutdowns must occur during the week and lastly whether a total shutdown of the Pittsfield Plant might be required.

In general, the major work contemplated for the August shutdown was the installation of valving on existing lines. The thought behind this was to permit carrying out new work or remaining work without any further main-line shutdowns. A very important factor bearing upon this scheduling was the effectiveness of the existing header valves on the power plant's 640 psi and 190 psi headers as well as on the 190 psi header bypass which fed various manufacturing areas.

These valves had not been used for some time and their ability to hold tight was an important consideration. For example if the existing valve on the 640 psi header did not hold then all 190 psi steam, all turbines, all boilers, and all manufacturing steam had

to be shut down to install the new 640 psi header dividing valve. Since this new valve with its stub pieces required stress relieving after welding, a 24-hour shutdown would not be enough and a full week-end would be needed. Similarly the other valves presented problems of their own.

After checking on these various valves it was agreed to shut the powerhouse down from August 7 to August 14. The piping work progressed from this point pretty generally on schedule. The early plans for the condensate piping made back in March were being put into effect by the contractor by August 20.

Boiler and Structural Steel

Structural steel erection got underway July 19 and the boiler supporting steel construction was all up by August 12 with the riveting 50 per cent completed. At this juncture the structural steel and the boiler erection work had to be dovetailed. For example, when the boiler drums began to go into place around August 20 the structural steel contractor had to handle his operations intermittently with those involved in the rigging and erection of boiler drums. By August 26 drum erection was complete and the air preheater installation already started. The structural steel to the fan floor and main roof level was then nearing completion and the penthouse framing had begun.

When September 17 rolled around the structural steel erection was finished, the steam generating tubes were almost all erected and expanded and the top and bottom sidewall headers in place. The dust collector shell and hoppers were in as was the air preheater and the base ring for the stack. Boiler feedwater header piping was moving along and drainage piping scheduled for installation beginning Septenber 14. October 1 was set for the start of the coal bunker erection and October 4 for the beginning of the stack assembly.

By October 1 work was far enough along to schedule the boiler hydrostatic test for October 16. The boiler waterwall and recirculating tubes were expected to be finished October 6. The piping for the hydrostatic test was to be shipped October 8. Blowdown piping, drain piping and valves were to be ready well in advance of October 16. The fans had been unloaded September 28; their foundations finished September 29 and were to be erected by October 4.

A successful hydrostatic test was performed October 20. Within roughly a month later the application of the plastic refractory to the boiler was well advanced. The skin casing was continuing well and the wall insulation and upper drum refractory protection was completed. The duct work was well along, stack guniting progressing, all remaining piping, pumping and control stations nearing completion by November 25. The boiler setting was then scheduled to be ready for drying out, December 6, with boil out and washing down to be completed December 10.

The preliminary operation of the FD and ID fans and of the air preheater was held December 2. Welding and stress relieving of the main steam piping was all finished by this latter date except for the joints that were to be made up December 11, 12, and 13 when the lead to one of the turbines was scheduled for cut-in. All instrument piping was held to be far enough along so that the boiler drying and boiling out operations could be carried out as planned.

Checking and connecting of electrical control circuits between the boiler gauge board and switchgear were completed as of December 2, The phase rotation of the motors driving the pulverizers, and the ID and FD fans, the chemical feed pumps, the clinker grinders and air preheater had been checked out and for the remaining motors was underway by this same date, The moving, checking and installing of the necessary switching equipment for the 10-in. motor-operated steam lead valve were reported as in progress as of December 2.

The last of the many weekly meetings was held December 9 with representatives from General Electric Company, the United Engineers and Constructors, Inc., the boiler manufacturer, Combustion Engineering, Inc., the electrical contractor, Collins Electric Company and the piping constructor, Hartwell Company. The minutes of this meeting are repeated here.

Boilers and Auxiliaries—

"Installation of doors completed, and of skin casing substantially completed. Refractory work finished December 7. Erection of all gas and air ductwork completed December 6. Running in of fans and air preheater occurred December 4 and 5. Erection of coal piping underway. Roughed-in insulation of air duct, gas duct, fans and superheater assembly nearly finished. Drying out and boiling out of boiler commenced December 7 at 9:45 p.m. Installation of retractable sootblowers to be completed December 14. Erection of sootblower piping to be completed December 17. Washing out of boiler scheduled December 11 firing boiler for blowing clean main steam piping and for setting safety valves scheduled for December 12. Boiler scheduled to be ready for steaming at operating pressure and low output December 14.

Piping and Mechanical—

"Erection of service water and drains nearly finished. Instrument piping substantially completed. Installation of ash removal piping and equipment to be finished December 10. Completion of boiler lead and No. 8 turbine lead, including connection to turbine, blowing clean and testing, scheduled for December 11, 12, 13.

Electrical Work—

"All electrical operating equipment connected and operable except coal feeder motors. Controls for ash handling approximately 85 per cent completed. Flame indicators to be installed the week of December 13. Installation of all lighting substantially completed. Installation of power and control for unit heaters underway and awaiting delivery and erection of heaters."

Conclusion

On December 14 the boiler actually went on the line at low outputs. About December 21 full rated load was placed on the boiler and from that day on the unit was a producing member of the powerhouse, Fig. 4.

The above description, of necessity, covered only the high spots. But it is evident that a well-detailed powerhouse construction program involving cooperative contractors and subcontractors under good supervision can achieve an erection and building schedule that on paper seems impossible.

*The Effect of Decreasing Boiler Pressure on Natural Circulation in Water Tube Boilers**

By O. G. HAMMAR and R. JUNG

A drop in boiler pressure occurring under load impacts can cause evaporation of water in downtakes. It has generally been assumed that this disturbs the circulating flow and endangers the heated riser tubes. A test unit representing a full sized section of a peak load steam generator was subjected to sudden load changes comparable to those in actual operation. Changes of circulation velocities in relation to time were measured.

It was found that the limiting value of the pressure-time gradient which will avoid disturbing circulation depends primarily upon the initial boiler pressure and the water velocity in the downtakes. If the latter is high and the riser tubes are heated at their lower part, circulation will be less affected. Restricting circulating flow by orifices in the downtakes will more easily lead to disturbances.

THE relationship between pressure-drop gradient and circulation is especially important for steam generators which are operated at full pressure and low load and are expected to deliver full load in response to rapid demand. Since the heat input cannot immediately follow a sudden demand, the increased heat requirements are at first met by pressure drop in the system. If circulation is to be maintained and the heated riser tubes are to be protected against damage by overheating, the pressure-drop gradient should not exceed a value which depends on the dimensions and arrangement of the circulating system, on the pressure and the initial circulating velocity. This limit determines the storage capacity available to meet sudden load demands.

The peak-load steam generator for which the investigation was to be made is shown in Fig. 1. A small traveling grate stoker keeps the unit at operating pressure (1120 psig) when steam is not required. The unit is to be put on the line at its full capacity of 375,000 lb per hr within less than one minute. The unit reacts to this impact with a pressure drop of approximately 425 psi in 150 seconds. By this time the oil burners have been started and can balance the pressure within another 30 seconds. Flashing of water in the downtakes b can be expected to occur, as shown below, as soon as the pressure gradient exceeds 60 psi per minute under the existing conditions. A required drop of 170 psi per min without disruption of circulation by flashing could then only be met by further increasing the storage capacity of a unit already equipped with two upper drums. The results of the tests undertaken to determine the actual limit made such measures unnecessary.

Theoretical Analysis

The arrangement of the natural circulation system is shown schematically in Fig. 2. The circulation moves in the direction of the arrow due to the difference in density between the water in the downtake b and the steam-water mixture in the heated riser d. The steam generated in d is separated from the water in the drum a and flows through the superheater c and the valve f to the turbine. As long as the steam generated equals the steam leaving the drum the drum pressure P remains constant.

This equilibrium is disturbed when valve f is suddenly opened and more steam leaves the drum. This requirement is at first met by flashing of water in line with decreasing boiler pressure. The difference between the heat in steam leaving the drum Q and the heat input Q_f

$$\Delta Q = Q - Q_f \qquad (1)$$

is removed from the system, the temperature of which will decrease by

$$\Delta\varphi/\Delta t = -\Delta Q/W \qquad (2)$$

where t is the time elapsed and W the storage value (Btu per deg F) of the boiler content taking part in the flashing process. The heat available due to the pressure drop of the steam contained in the drum can be neglected. The process of steam flashing follows the steam pressure line indicating the connection between saturation pressure P and saturation temperature φ. In the purely formal relationship

$$\Delta P/\Delta t = (\Delta P/\Delta\varphi)\,(\Delta\varphi/\Delta t) \qquad (3)$$

the ratio $\Delta P/\Delta\varphi$ represents the average gradient of this process between its beginning and ending.

The storage value which can be considered a design characteristic can then be found from equation (2) and (3) as

$$W = -\frac{\Delta P/\Delta\varphi}{\Delta P/\Delta t}\,\Delta Q \qquad (4)$$

* Translated by W. W. Schroedter, Combustion Engineering, Inc., from an article which appears in *Brennstoff-Wärme-Kraft*, Vol. 7, No. 1, January, 1955.

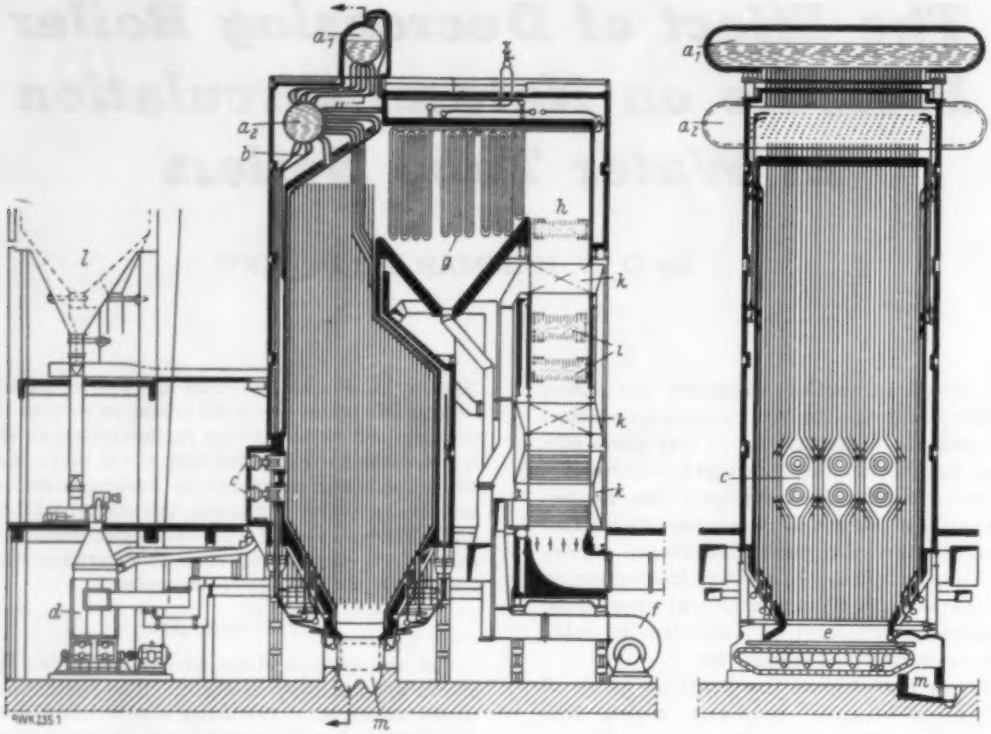


Fig. 1.—Steam generator for 375,000 lb per hr, 1140 psig, 932 F with combined oil and pulverized coal firing and traveling grate stoker for standby operation

(a_1) Dry drum
(a_2) Steam drum
(b) Downtakes
(c) Six burners for oil and pulverized coal
(d) Three pulverizers
(e) Traveling-grate stoker
(f) Two f-d fans
(g) Superheater
(h) Evaporating surface
(i) Economizers
(k) Airheater
(l) Coal bunker
(m) Ash sluice

in which the heat differential ΔQ is given and the pressure-time gradient $\Delta P/\Delta t$ should be below the disturbance limit established in this article.

When the pressure drops below the saturation temperature of the water in the drum, flashing of steam occurs first in the drum and riser tubes, where a saturated condition always exits. If the pressure-time gradient exceeds a certain limit, then steam forms in the downtakes as well and circulation can, under conditions yet to be determined, be reduced and even stopped. This means danger of local overheating of risers not properly cooled and consequently tube failures.

The pressure-time gradient which will limit flashing in downtakes is determinable through the condition that the drop of drum pressure must equal the pressure increase to which a water particle would be subjected at constant drum pressure in the downward flow by reducing its elevation. This pressure increase is

$$\Delta P_F = \gamma' \Delta h - \Delta P_R \tag{5}$$

where γ' represents the water density at saturation temperature φ_0, Δh the change in elevation and ΔP_R the pressure drop by friction. This pressure drop occurs along the flow path

$$\Delta x = \Delta h/\cos \alpha \tag{6}$$

where α is the angle between the tube axis and the vertical. To cover this distance the water particle which has a velocity w_F, needs a time interval of $\Delta t = \Delta x/w_F$ and is, according to equations (5) and (6), subject to a pressure increase

$$\Delta P_F = w_F \gamma' \Delta t \cos \alpha - \Delta P_R \tag{7}$$

The conditions for the water particle considered do not change if the drum pressure decreases by $\Delta P = \Delta P_F$ within the interval Δt. The water flow in the downtake, initially ($x = 0$) saturated, will not contribute to flashing when

$$-\Delta P/\Delta t \leqq w_F \gamma' H/L - \Delta P_R/\Delta t \tag{8}$$

where cos α from equation (7) was replaced by the ratio of the height H to the tube length L (Fig. 2).

The friction part of the pressure gradient is determined by

$$\Delta P_R/\Delta t = \lambda(w_F/d_F)w^2{}_F\gamma'/2g \tag{9}$$

with λ as friction coefficient, also taking into account losses at the tube entrance and in elbows.

The conditions under which the system shown in Fig. 1 avoids local flashing are presented in Fig. 3 where the length coordinate x (Fig. 2) is plotted against the elevation differential h. Because of the relationship $\Delta P = \gamma' \Delta h$ this then represents the distribution of static pressure in the downtake at the no-flow condition. The downtake runs vertically between the points 0 . . . 1, 2 . . . 3, 4 . . . 5, 6 . . . 7, and 8 . . . 9 ($\Delta h/\Delta x = 1$). The loop shown in Fig. 2 lies between the points 3 and 6. The straight connection between the end of the down-

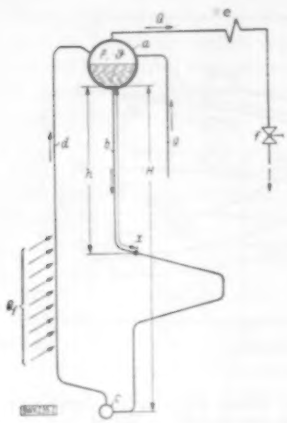


Fig. 2.—Schematic arrangement of a natural-circulation system

(a) Drum
(b) Downtake
(c) Header
(h) Riser
(e) Superheater
(f) Steam valve
(g) Feedline

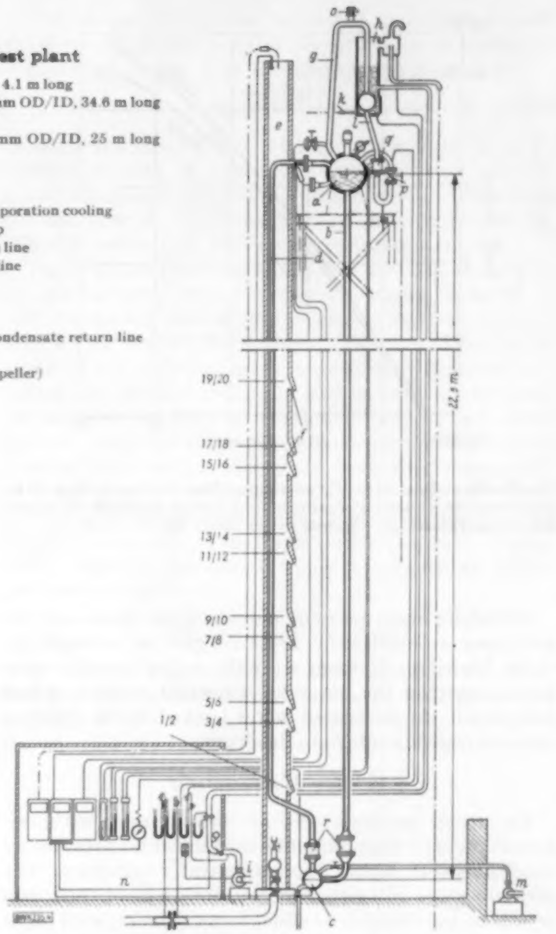


Fig. 4—Test plant

(a) Drum, 910 mm ID, 4.1 m long
(b) Downtake, 83/73 mm OD/ID, 34.6 m long
(c) Header
(d) Four risers, 70/60 mm OD/ID, 25 m long
(e) Stack
(f) 20 gas burners
(g) 10 steam pipes
(h) Condenser with evaporation cooling
(i) Cooling water pump
(k) Cooling water drain line
(l) Condensate return line
(m) Feedpump
(n) Test room
(o) Blowoff valve
(p) Throttle valve in condensate return line
(q) Venturi meter
(r) Velocity meter (propeller)

take, shown by the dotted line between points 0 and 10, represents the limit value of equation (8) where the friction loss is neglected. When this line cuts the static pressure line between I and II, a very small amount of water evaporates which does not influence the flow and which has condensed in reaching point 10.

The limiting pressure gradients are calculated for this example where $H/L = 0.65$ and $\gamma' = 740$ kg/m³ (at 70 ata) according to equation (8)

for	$w_F = 1$	2	3	m/sec
to	$-\Delta P/\Delta t = 2.85$	5.7	8.5	at/min

Frictional pressure drop reduces these values by

2.7 11 25%.

When the unit under investigation is in a standby condition, the water velocity in the downtakes amounts to approximately w_F = 1.6 m/sec. Flashing will then occur in the downtakes when the pressure-time gradient exceeds 4.2 at/min. or 60 psi/min.

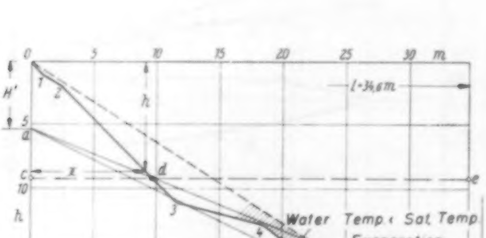


Fig. 3—Arrangement diagram of downtake of test plant, length versus height

It has been suggested (1) that flashing can be avoided by adding water with lower temperature into the downtakes. These and similar measures such as outside cooling of downtakes add considerable design and operational complications which our test results do not justify.

Test Model

The circulating system used for this investigation was built as a section of the proposed standby generator in full size in order to remove any doubt as to the applicability of the results. The system with its main dimensions shown in Fig. 4 consists of the drum *a*, the downtake tube *b*, the lower header *c* and the four risers *d* which are heated in a stack *e* with a maximum of 20 gas burners (in two parallel rows, No. 1 to 20).

In the standby condition the generated steam is removed from the drum by 10 tubes *g* to the condenser *h* which is cooled by evaporating water at atmospheric pressure. The condensation is adjusted to the evaporation in the boiler by raising or lowering the cooling water level in the open tank *h* with the cooling water pump *i* or the drain pipe *k*. The condensate forming in the tubes *g* returns into the drum through the line *l* which contains a loop with a venturi *q* for metering the condensate flow. The system is supplied by the pump *m*.

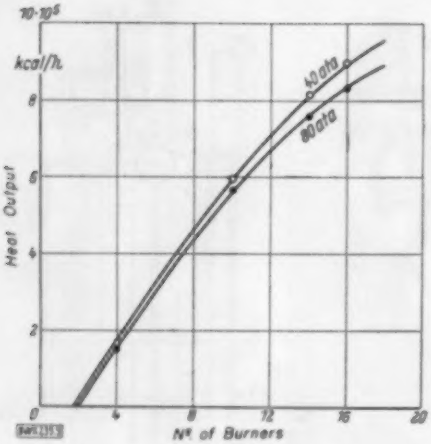


Fig. 5—Heat flow $Q = Dr$ corresponding to steam flow D to condenser (r = heat of evaporation) versus number of burners in operation, n. Lower firing limit $H_o = 0.1\ H$

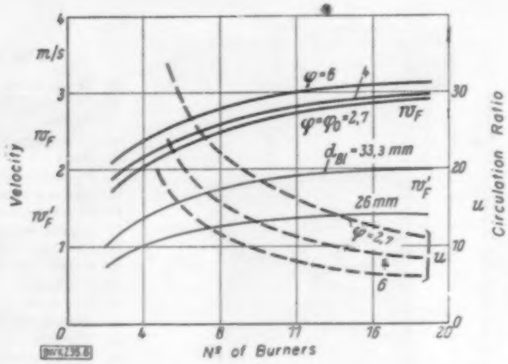


Fig. 6—Relationship between water velocity in downtake and heat input at 80 ata and $H_o = 0.1\ H$

w_F' = Measured velocity at throttled circulation
w_F = Velocity for area ratio corresponding to throttle condition
ϕ = Area ratio: risers/downtakes
u = Circulating ratio
d_{Bl} = Diameter of orifice

All tubing under pressure outside of the stack and the condenser is insulated. Pressure can be reduced by either blowing off steam with the valve o or by completely opening the otherwise throttled valve p which will permit the subcooled water backed up in the condenser to quickly return to the drum.

Velocity Measurements

The usual methods of flow measurement with orifices or pressure taps cannot be used under non-stationary conditions with highly fluctuating water velocities. The pressure gages will either reflect vibrations of their own or will be too sluggish to allow correct timing with other measurements. A special device proposed by R. Burgholz (2) and further developed by the TÜV Essen (3, 4) was therefore used. A low-friction propeller centered inside the tube carries an isotope in a pressure-tight capsule on one of its blades. Every revolution will activate a Geiger counter connected to an oscillograph outside of the tube. One of these instruments r is used for each riser and for the downtake (Fig. 4).

Tests with Constant Load

When the boiler load is kept constant the circulation depends on the heat input and its distribution over the length of the risers, on the pressure and the physical arrangement of the system. The distribution of the heat input is varied by increasing, reducing or throttling, the gas burners and is indicated for each test by the evaluation of H_o of the lowest burner operated above the lower header.

At maximum heat input approximately 10^6 kcal/hr were absorbed by the four risers with a peak absorption rate of 200,000 kcal/m² hr occurring at the upper burners. The low steam release rate in the drum gave assurance of dry saturated steam entering the tubes g (Fig. 4). Their pressure drop and that of the condensate return line l was small enough to keep the condensate level in l below the cooling water tank h. Subcooling of condensate was therefore restricted to less than 5 deg C for all stationary tests. The heat output Q calculated from the condensate flow K measured at q and the latent heat r to $Q = Kr$ can therefore be set equal to the heat input minus the heat losses of the system.

The heat output Q is plotted in Fig. 5 against the number of burners, starting with the lowest position. Only the minimum distance $H_o = 0.10\ H$ between the lower header and the lowest burner remains unheated. The two curves measured for boiler pressures of 80 ata and 40 ata show that two burners can maintain the standby condition ($Q = 0$).

The relationship between number of burners and water velocity in the downtake is shown in Fig. 6 for three throttle conditions of the circulation system in parameters ϕ, which originally represented the ratio of total flow areas of riser tubes and downtakes.

$$\phi = df_a/bf_b \tag{10}$$

For the circulation system in Fig. 4 this ratio without throttling is $\phi = \phi_0 = 2.70$. To test the effect of an

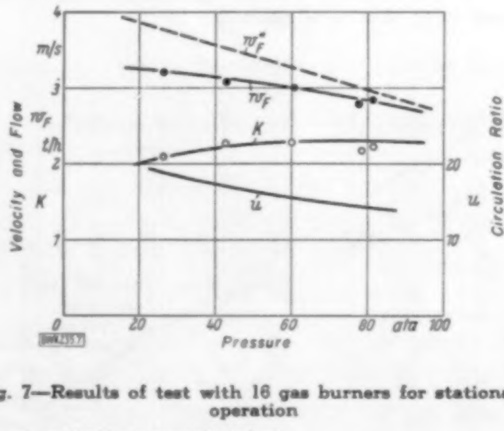


Fig. 7—Results of test with 16 gas burners for stationary operation

w_F = Velocity in downtake (m/sec)
w^*_F = Velocity in downtake, calculated according to equation (5).
K = Condensate flow (t/h)
P = Pressure (ata)

increase in this ratio on circulation it was merely necessary to insert an orifice into the downtakes. This increased the tube resistance based on the velocity head of the downtake flow $q_F = w^2_F\, \gamma'/2g$ from its initial value ξ_0 to

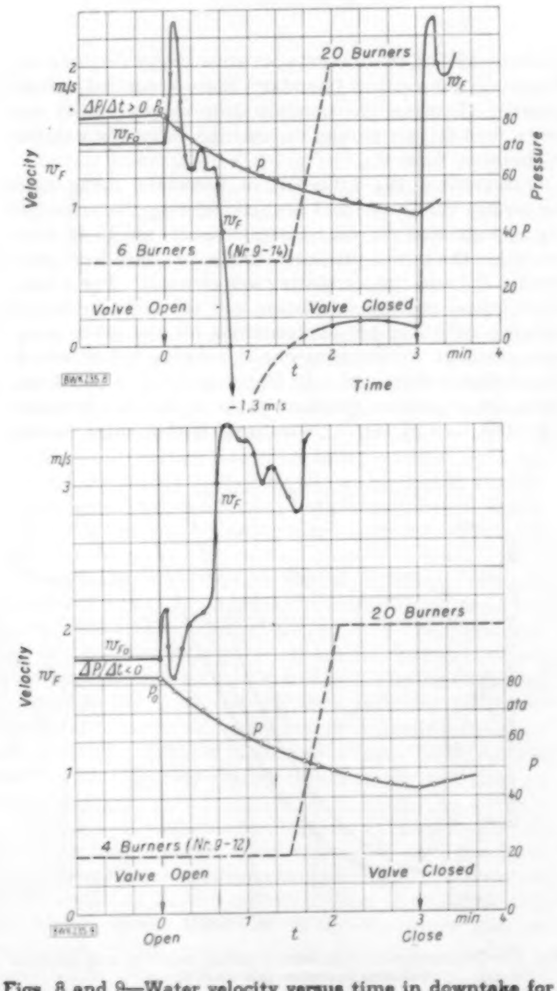
$$\xi = \xi_0(\phi/\phi_0)^2 \tag{11}$$

The circulating ratio (ratio of water flow to steam generated D)

$$u = w_F \cdot F\gamma'/D \tag{12}$$

(where F is downtake area) for equal absorption rate is then as large as in an unthrottled system in which the same value of ϕ was achieved by a different ratio of d/b with tubes of the same dimensions.

The curves of the circulating ratios u in Fig. 6 were calculated from the measured values of the water velocity w_F and the steam flow D ($= Q/r$, Fig. 5). They show u decreasing with more throttling. The steam ratio in the circulating flow then increases which favors separation in the drum and can give better steam purity for high release rates.



Figs. 8 and 9—Water velocity versus time in downtake for different initial velocities w_{F_0} and equal pressure drop. For test on Fig. 8, circulation stops ($w_F \to 0$, disturbance)

The water velocity w_F' measured in throttled downtakes corresponds to a water velocity w_F in an equivalent unthrottled system with the area ratio ϕ is equal to

$$w_F = w_F'\, \phi/\phi_0 \tag{13}$$

When ϕ is increased by a change of the ratio d/b, the average density of the steam-water mixture also decreases and makes available a larger pressure differential for the flow in the downtake. This explains the increase in velocity w_F with an increasing area ratio ϕ.

Fig. 7 shows the results of a test series working with 16 gas burners. The curve of circulation velocity w_F and condensate flow K was measured at constant pressure for an unthrottled system. The dotted line is the result of a calculation made according to H. Seidel and, basing the steam bubble slip and flashing in the risers, on proposals made by E. Schmidt (6) and on test values from P. Behringer (7). For simplification it was further assumed that the heat absorption was in each case equally distributed along the length of the tubes with two-thirds between the lowest burner and a point midway in the stack and one third in the upper half of the stack. Agreement with the measured values especially at higher pressures is satisfactory.

Tests with Variable Load

In running tests with decreasing pressure, the firing rate and the operating procedure were used to control the initial value of the circulation velocity w_F while the position of the relief valve o (Fig. 4) regulated the

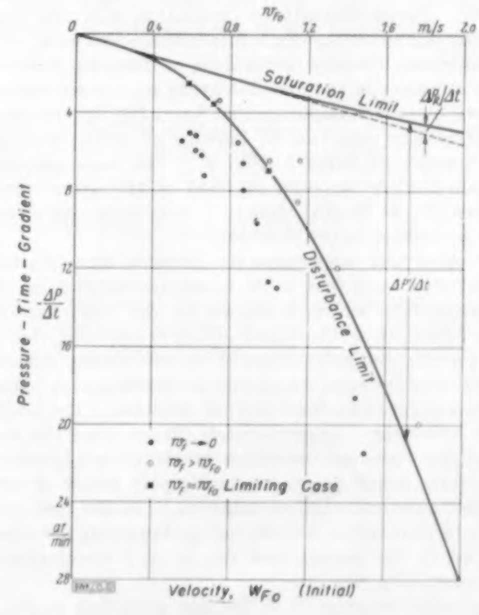


Fig. 10—Disturbance diagram for circulation at initial pressure of 80 ata

All initial conditions below the disturbance ($w_F \to 0$)
Limit curve lead to stoppage of circulation. Instable boundary conditions are on limit line.
Circulation accelerates for conditions above limit line: $w_F > w_{F_0}$
Saturation limit divides area of steam free water flow in downtake from area below where flashing occurs.
The influence of resistance due to friction $\Delta P_R/\Delta t$ is noticeable above $w_{F_0} =$ 1.2 m/sec.

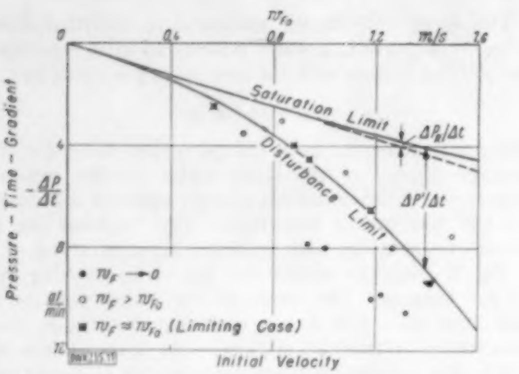


Fig. 11—Disturbance diagram of circulation for initial pressure of 50 ata. (Compare Fig. 10)

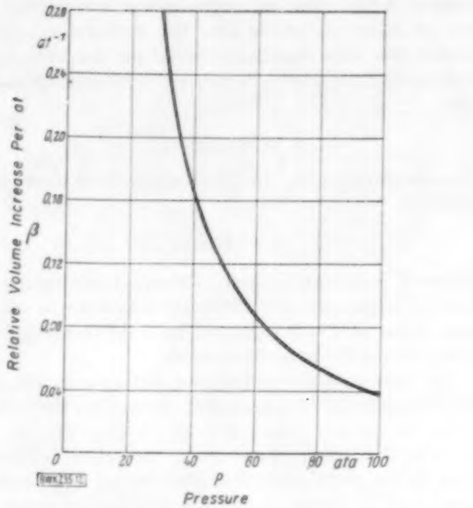


Fig. 12—Relative volume Increase $\beta = \Delta v/v'\Delta$ P of saturated water for a pressure drip of P = 1 at versus pressure. (From steam tables)

pressure-time gradient. Because of their wide fluctuation during the process of dropping pressure, it was considered unnecessary to establish an initial stationary condition with constant pressure and circulation velocity. This allowed a considerable reduction in testing time because the circulating velocity, for instance, could be easily corrected at constant heat input by increasing or decreasing the level of cooling water in the condenser. The resultant change in boiler pressure (approximately 0.5 at/min max) has no practical influence on the behavior of circulation when the load impact occurs.

For some tests boiler pressure was dropped by opening valve p (Fig. 4) after having backed up cooled condensate. Better stability of circulation was observed, probably due to the sudden influx of subcooled water into the downtake, thereby eliminating or delaying flashing.

The influence of the initial velocity w_{F_o} on circulation is shown by two examples in Fig. 8 and Fig. 9. Pressure was reduced in each case by blowing off steam with the same opening of valve o (Fig. 4). The same pressure gradient ($\Delta P/\Delta t = -20$ at/min) at the same initial pressure ($P_o = 80$ ata) showed a wide disparity of circulation velocity in the downtake.

For these and other tests the pressure drop starting at the time $t = o$ first leads to an acceleration of the circulating flow which is caused by the instantaneous start of flashing in the risers. This is followed, due to inertia forces, by fluctuations of the circulating velocity in a period of the same magnitude as the frequency period of a communicating fluid column with twice the length of the downtake. Approximately 30 sec after the load impact the velocity moves towards a new state of equilibrium which is either considerably below or considerably above the initial value w_{F_o}. In the first case ($w_F \rightarrow 0$) circulation is disturbed endangering the risers (Fig. 8); in the second case ($w_F > w_{F_o}$) circulation is maintained (Fig. 9).

The initial pressure P_o in the test according to Fig. 8 was reached with a slight increase in pressure ($\Delta P/\Delta t > 0$) and six burners. The other burners were lighted 90 sec after opening the relief valve. This change in firing does not show any influence on the velocity behavior. In some instances it was found, however, that the increase in firing during lower pressure-time gradients can restore a previously disturbed circulation to satisfactory levels. As to their initial velocity, w_{F_o} and pressure gradient $\Delta P/\Delta t$, these events occurred under similar conditions as for certain boundary cases described subsequently. In these the pressure drop merely causes velocity fluctuations around an average value comparable to the initial value w_{F_o}.

Disregarding the influence of increased firing rate for certain conditions and first considering the same firing arrangement H_o, only three variables are to be considered: the initial pressure P_o, the pressure-time gradient $\Delta P/t$ and the circulating velocity w_{F_o}. For a constant initial pressure a relation can therefore be found between $\Delta P/\Delta t$ and w_{F_o} determining for any given pressure gradient a minimum initial velocity below which circulation is disrupted. In Fig. 8 and Fig. 9 this limit value for a pressure gradient of 20 at/min lies between 1.45 and 1.78 m/sec. More tests within these values

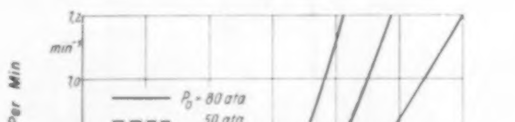
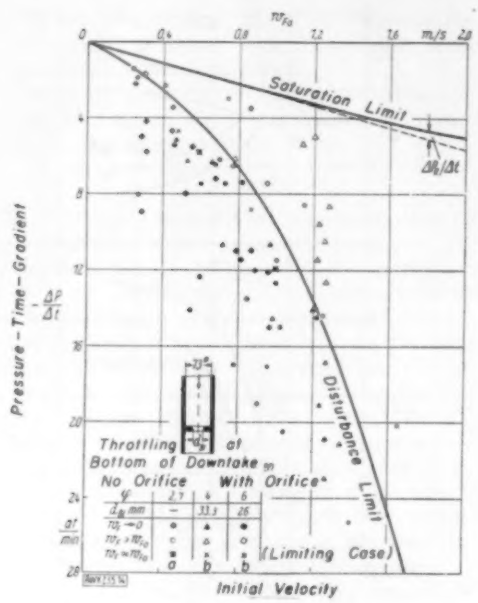
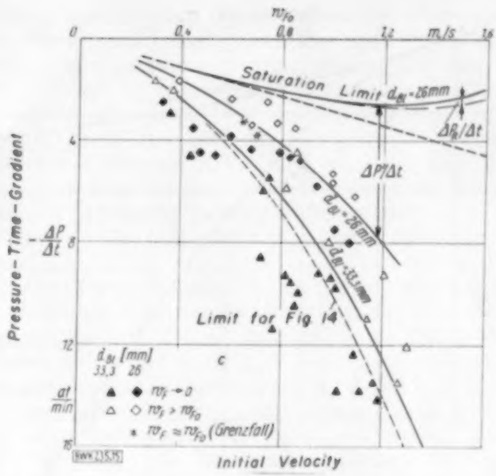


Fig. 13—Disturbance limit versus initial velocity and relative volume increase per unit time

d_F Downtake ID
H_o Lower firing limit





Figs. 14 and 15—Disturbance diagram at 80 ata and lower firing limit. $H_e = 0.1\,H$; downtake 73 mm ID

(a) System not throttled
(b) System throttled at lower part of downtake
(c) System throttled at downtake inlet

will further narrow down the area of expected disturbance. The results for a variation of initial conditions ($\Delta P/\Delta t$ and w_{F_0}) are shown in Figs. 10 and 11.

The test points plotted in Fig. 10 for an initial pressure of approximately 80 ata show the initial conditions of velocity and pressure gradient and by their symbols the circulation behavior in each case. The lower curve divides the areas of disturbed and maintained circulation. Any relationship of initial conditions below the limit line leads to a circulation disturbance according to Fig. 8. For conditions above the line the load impact accelerates circulation, as shown in Fig. 9. The conditions which will cause a fluctuation of velocity about the initial value are found on the limit line. The upper curve with an initial gradient of $-(\Delta P/\Delta t)/w_{F_0} = \gamma' H/L$ describes the limiting condition as represented by equation 8. At higher velocities the curve bends slightly upwards due to the increasing influence of friction losses. Above this line there is no flashing of water in the downtake.

These results show that circulation, especially for larger initial velocities, is surprisingly insensible to load impacts. The requirement of $\Delta P/\Delta t = -12$ at/min and $w_{F_0} = 1.6$ m/sec and $P_e = 80$ ata for instance, which is an extreme usually not found in the boiler industry, is found in Fig. 10 to be safe with sufficient tolerance. It will take a pressure gradient of 18 at/min for the same initial velocity to cause a circulation disturbance despite the fact that flashing already occurs in the downtake at 4.2 at/min.

This type of disturbance diagram is shown in Fig. 11 for an initial pressure of 50 ata. Although the limit curve has the same tendency as in Fig. 10 for a pressure of $P_e = 80$ ata, it is considerably less steep, especially for higher initial velocities. This divergence can be explained by the results of the following calculation based on the supposition that circulation flow at the disturbance limit will behave similarly at a different initial pressure P_e when at the same initial velocities w_{F_0}. A similar flow under these conditions, however, can only be expected if the pressure gradients $\Delta P/\Delta t$ lead to the same change of specific volume in relation to time when flashing occurs.

With x as the then developing steam ratio, v' the specific volume of saturated water and v'' the specific volume of the saturated dry steam, the increase in volume is

$$\Delta v = x(v'' - v') \tag{14}$$

Since the process is nearly isentropic, the relationship between x and the pressure drop ΔP is given by

$$rx = (di'/dP - v')\Delta P$$

or in sufficiently close approximation

$$rx = (di'/dP)\Delta P$$

where di' represents the enthalpy change of the water, at the saturation limit for the pressure change dP and r is the heat of evaporation. Using the steam ratio as

$$x = \frac{(di'/dP)}{r}\,\Delta P \tag{15}$$

equation (14) provides the relative volume increase to

$$\Delta v/v' = \frac{(di'/dP)}{r}\cdot\frac{v'' - v'}{v'}\,\Delta P \tag{16}$$

The property

$$\beta = \frac{\Delta v/v'}{\Delta P} \tag{17}$$

can be calculated from the steam tables as shown in Fig. 12 as a function of P and used to transform the disturbance limits found for 50 and 80 ata. It should be noted that only the pressure differential $\Delta P'/\Delta t$ (Figs. 10 and 11) between saturation limit and disturbance limit causes flashing. If the limit is only determined by w_{F_0} and the corresponding relative volume increase

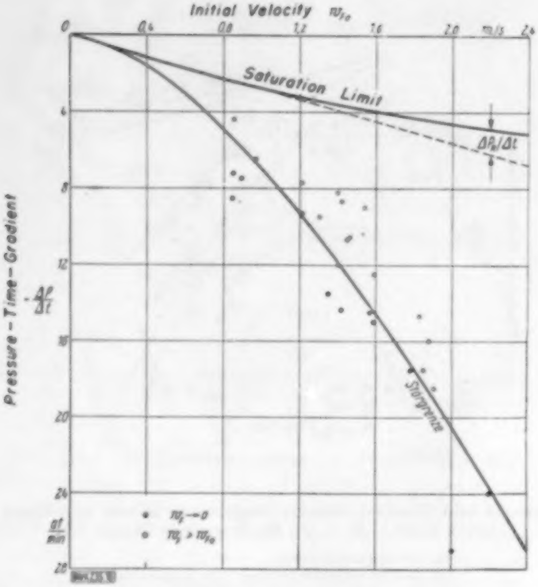


Fig. 16—Disturbance diagram at 80 ata and H_o = 0.1 H; downtake 47 mm ID

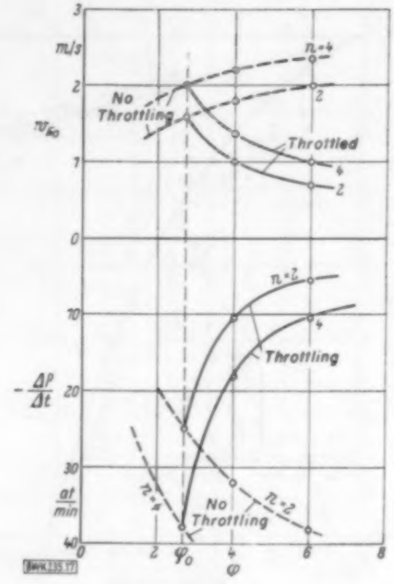


Fig. 17—Water velocity in downtake w_{F_o} and pressure-time gradient for equal heat input with n = 2 and n = 4 burners, versus area ratio ϕ and corresponding throttling which produces for stationary conditions the same circulating ratio (P_o = 80 ata, H_o = 0.1 H)

$$\frac{\Delta v/v'}{\Delta t} = -\beta(\Delta P'/\Delta t) \tag{18}$$

then a relationship independent of the initial pressure P_o should be expected between $(\Delta v/v')\Delta t$ and w_{F_o} at the disturbance limit. This does happen as shown by Fig. 13 where the two disturbance limits corresponding to Figs. 10 and 11 coincide. For any values of initial velocity and initial pressure the maximum allowable pressure-time gradient can therefore be presented as

$$-\Delta P/\Delta t = (1/\beta)\,(\Delta v/v')/\Delta t + (w_{F0}\,\gamma' H/L - \Delta P_R/\Delta t) \tag{19}$$

The relative volume increase $(\Delta v/v')\Delta t$ is determined in this equation by w_{F_o} in Fig. 13 and the property β by P_o in Fig. 12. The final term is the limiting value of the pressure-time gradient for water in the downtake without steam according to equation (8).

The independence of the disturbance limit from the initial pressure P_o would indicate that within safe limits the steam-water mixture in the downtakes behaves as a homogeneous fluid in which the diffusely distributed small steam bubbles have no relative movement. In sharp tube bends and for low initial pressure with large density differences between water and steam, a separation of both phases producing a circulation disturbance can be expected above the limit line shown in Fig. 13. The results of these tests can therefore be applied only to similar circulating systems and to pressures above approximately 30 ata.

Special Conditions

The circulation behavior under decreasing pressure is not only influenced by the values of P_o, $\Delta P/\Delta t$ and w_{F_o}, but also by the distribution of the heat input along the risers. This influence is actually characterized only by the lower firing limit H_o where flashing begins determining the steam distribution during the first phase of the event. The disturbance limits in Figs. 10 and 11 are drawn for an initial firing with the middle burners (No. 9 . . .) which places the lower firing limit at H_o = 7 m or ≈ 0.3 H above the lower header.

Test results run with an initial firing of the lower burners (No. 1 . . . , H_o = 0.1 H) are shown in Fig. 14. The diagram also shows the test points from the throttled circulation system for the same firing arrangement. The orifices in this case were installed between two flanges in the downtake about 2 m above the lower header.

As shown by Fig. 14, all tests points of the three systems indicate the same disturbance limit. As far as circulation at that limit is concerned, the opposing effects of friction resistance and initial firing level appear to be widely in balance. For the same initial velocity the increased resistance of the throttled system corresponds to the increased initial firing which maintains its influence on the steam distribution in the risers and therefore on the circulation well into the decisive phase of the pressure drop. Because flashing already starts 2 m above the lower header, the disturbance limit curve runs steeper than for the middle firing level (H_o = 7 m, Fig. 10), especially for initial velocities below w_{F_o} = 0.6 m/sec. Transferring this limit to the relation $(\Delta v/v')/\Delta t$, w_{F_o} produces the upper curve in Fig. 13.

The pressure drop at the orifice causes additional flashing in the lower part of the downtake which is effective only on a short path ahead of the risers and has therefore practically no influence on the density of the steam-water mixture in the downtake averaged over the height H. The steam generated at an orifice placed in the upper part of the downtake, however, will flow

through the whole height H. This displaces the disturbance limit, as shown with further test results in Fig. 15, into a previously safe area.

In order to estimate the saturation limit for this case, the zero axis in Fig. 3 is first lowered by the head due to the pressure drop at the orifice:

$$H' = \xi_{B1} w^2_{F0}/2g$$

In Fig. 3 the curve obtained with the discontinuity H' at the starting point describes the static pressure distribution in the downtake. The inclination of the straight line $(a - 10)$, which is $(H - H')/L$ and is to be substituted for the expression H/L in equation (8), determines the limiting pressure gradient $\Delta P/\Delta t$ for the following condition. The pressure of a fluid particle at the end of the downtake equals the initial pressure at the beginning of the downtake. The straight line $(a - 10)$ however runs below the curve of the pressure distribution and demonstrates that all of the downtake volume will participate in flashing at the limiting pressure gradient $\Delta P/\Delta t$ calculated with that inclination. The condition of equal pressures is therefore insufficient for determining the saturation limit.

Following equation 8 this limit was defined as determining the relationship between the pressure-time gradient and the velocity w_{F_0} for the condition that saturated water only enters the lower header. The saturation limit thereby corresponds to an average of the local static pressure increase $\Delta P/\Delta$ for $\Delta P/\Delta t = 0$ which is found in first approximation from the slope of the line $(a - b)$ in Fig. 3. The straight line has been drawn for $w_{F_0} = 1.5$ *m*/sec in such a manner that the areas $a - 0 - 1 - 2 - d - a$ and $d - b - 10 - 9 - 8 \ldots 3 - d$ are equal. It is hereby assumed that the steam generated along the path $x = c - d$ in the upper part of the downtake condenses again between d and e and that its influence on the pressure distribution can therefore be neglected. If the slope H/L in equation (8) is now replaced by slope δ of the straight line $(a - b)$ the equation of the saturation limit can be written as

$$-\Delta P/\Delta t = w_{F_0}\gamma'\delta - \Delta P_R/\Delta t$$

where δ depends on w_{F_0} because of the head H' and where the resistance term is identical with that of the unthrottled system. The saturation limit calculated in this manner is plotted in Fig. 15 for an orifice with a contraction ratio of $\phi = 6$. The curve considerably exceeds that of the unthrottled system especially for velocities above one m/sec, but fails to compensate for the change in the disturbance limit. Probably a separation occurs in the two-phase mixture behind the orifice making the circulation more liable to disturbances.

Additional tests were made with downtakes of 47 mm ID instead of the previously used 73 mm, as plotted in Fig. 16. The higher pressure drop due to increased resistance results in a more pronounced divergence of the saturation limit from the initial tangent which represents the limit for steam-free water at the end of the downtake without friction. The disturbance limit has now been raised as shown in Fig. 13 and shows a tendency similar to that of the throttled system described above.

Conclusions

The presentation of the disturbance limit as used above is not sufficient to judge behavior of similar circulating systems when the pressure drops. A comparison under equal outside conditions requires an equal amount of evaporation or firing prior to the load impact, but not an agreement as to circulating velocities w_{F_0}. Both the unthrottled system and the system throttled at the downtake end show in Fig. 14 the same relationship between $\Delta P/\Delta t$ and w_{F_0}. For the same heat input, however, throttling of the downtake leads to a smaller circulating velocity while an increase in the area ratio ϕ (Eq. 13) results in a higher downtake velocity.

For the latter condition, which can be imitated only for stationary operation by throttling, the assumption suggests itself that here, too, the same relationship between $\Delta P/\Delta t$ and w_{F_0} (Fig. 14) exists. This assumption and the relationship between w_F, ϕ and n of Fig. 6 were used to establish the comparison shown in Fig. 17.

Both the velocity in the downtake $w_F = w_{F_0}$ taken from Fig. 6 and the corresponding pressure-time gradient $\Delta P/\Delta t$ at the disturbance limit taken from Fig. 14 have been plotted versus the area ratio ϕ for the firing arrangement with 2 and 4 burners (Nos. 1 and 2, and Nos. 1 to 4, Fig. 4). A distinction should thereby be made between the area ratio of the unthrottled system as defined by equation (10) and the corresponding throttling which produces the same circulating ratio at constant pressure.

The curves which start at the area ratio $\phi = \phi_0$ of the unthrottled test system show an opposing tendency. Increasing the area ratio increases the allowable pressure-time gradient; throttling reduces it. The use of orifices to reduce circulation for improvement of steam dryness is therefore limited, not only due to an otherwise usually harmless reduction of the circulating ratio, but rather because of the increased liability to circulation disturbance under load impact.

The test results are applicable only to similar circulating systems with similar dimensions and equally good distribution of water to the risers. A report on tube failures due to circulation disturbances (8) shows that low-head boilers with downtakes connected to the front end of the lower headers are especially liable to trouble.

Note: The report was supported by funds from the Chalmerska Forskningsfonden and by the assistance of Kjell Delling, Malmo, and R. Semp, Goteborg. The Technischer Uberwachungsverein Essen participated in the tests in supplying the velocity measuring equipment. Dr. Ing. Schwarz, Dipl. Ing. Freude and Dipl. Ing. Theobald were active collaborators. The test plant was built and operated by L. C. Steinmuller, Gummersbach.

Note: Three units as shown on Fig. 1 were built by L. C. Steinmuller, Gummersbach for the Sydsvenska Kraftaktiebolaget, Malmo, Sweden.

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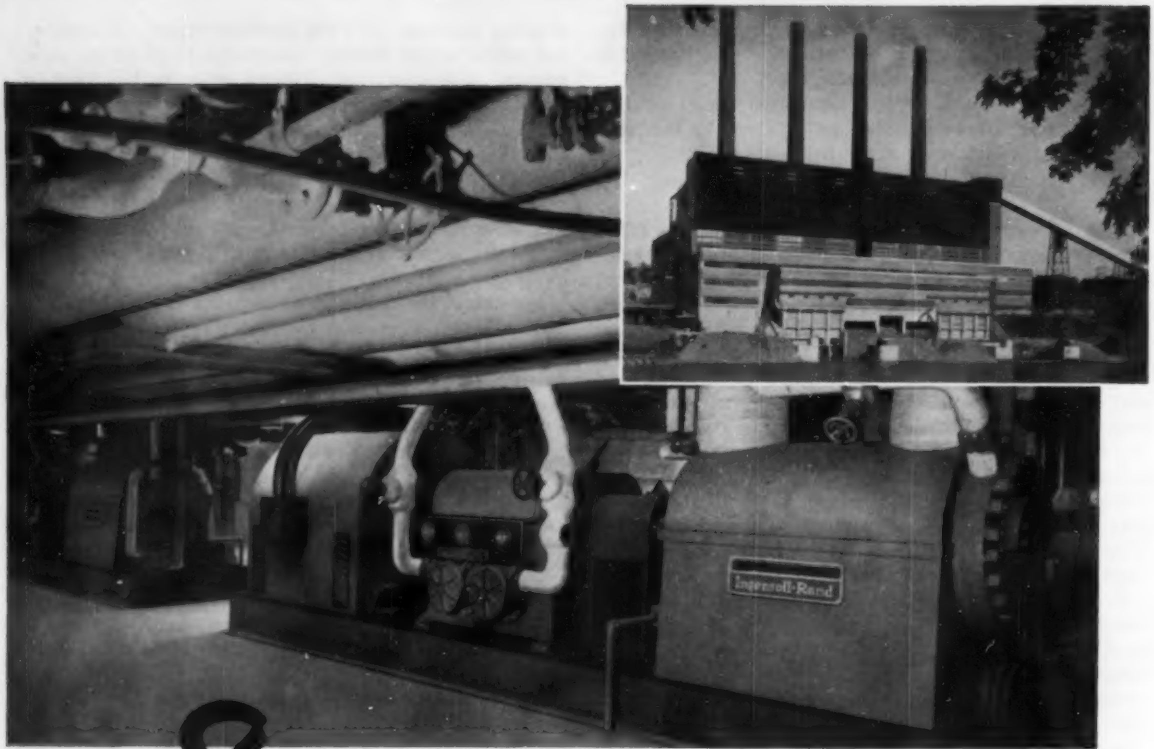
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Selection of Cooling Towers for Generating Stations*—II

By GEOFFREY F. KENNEDY, M.A.,† and P. H. MARGEN, B.Sc., (Eng.)‡

In this, the second of a two-part article, the authors suggest a way of framing cooling cooling tower specifications for generating station applications under various climatic and load conditions and for various turbine characteristics. Natural and mechanical draft towers are also compared.

THE basic procedure for determining the economic duty of a cooling tower serving a generating station is well known. It consists of evaluating: a—The additional annual investment in the cooling tower required to give a specific improvement in performance, e.g., a reduction in the inlet and outlet water temperatures by 1 degree F. at specified atmospheric and heat load conditions. b—The financial value of the improvement in performance to the generation authority, i.e., the value of the additional electricity which can be generated for the same station heat intake, less any increase in the cost of primary plant items such as the turbo generators. The economic duty point is that for which a = b.

The annual investment, a, can be evaluated from the performance equations 1a, 1b or 2 and cooling tower cost data. The value, b, depends mainly on the following three conditions, as shown in reference (2)[1]:

i The turbine characteristic
ii The annual load program
iii Climatic condition

Condition i is determined largely by the rate of heat rejection per sq ft of net turbine exhaust area, which may be termed the "exhaust heat loading," q_e. The greater q_e, the greater is the turbine leaving loss and hence the smaller the improvement in turbo generator efficiency per degree reduction in the vacuum temperature.

Condition ii is defined largely by the annual load factor, L, if a typical set of load curves is associated with each value of L as illustrated on Fig. 1.

Condition iii depends primarily on the annual mean wet bulb temperature and to a lesser extent on the seasonal variations in temperature. Table I shows typical seasonal variations applicable for the British climate and used in calculating most of the charts in this article.

For given assumptions concerning the secondary conditions such as the shape of the load curves for a given load factor, or the magnitude of seasonal deviations from the annual mean temperatures conditions i, ii and iii are thus fully determined by the turbine exhaust heat loading, the annual load factor, and the annual mean wet bulb temperature respectively, and the economic cooling tower conditions can be found for each combination of these three parameters.

TABLE I—MEAN ATMOSPHERIC TEMPERATURES*

	Season (4 Months Each) Winter	Intermediate	Summer
Wet bulb °F	41	49	57
Dry bulb °F	43	52	61

* Assumed for Figures 3, 4, 5.

Economic Duty Conditions

The economic solution for the cooling tower is affected also by the other plant items in the heat rejection circuit illustrated on Fig. 2, i.e., the condenser and the circulating water pump, and the economic rating of these three plant items must therefore be determined simultaneously for each set of conditions. The ratings are conveniently specified by stating the full load rate of heat rejection, the design atmospheric conditions and the following three temperature differences:

$(\theta_m - t_{w1})$ = the algebraic mean temperature difference across the cooling tower which is an inverse measure of cooling tower size, and where $\theta_m = (\theta_1 + \theta_2)/2$ and t_{w1} is atmospheric wet bulb temperature.

$(t - \theta_m)$ = the algebraic mean temperature difference across the condenser, which is an inverse measure of condenser size, and where t is the steam temperature in the condenser.

$(\theta_1 - \theta_2)$ = the temperature range of the circulating water which is an inverse measure of the water quantity, and also influences the cooling tower and condenser size to a minor extent.

Figs. 3 and 4 illustrate how the economic values of these three parameters can be represented graphically for a given climate for all combinations of load factor and turbine exhaust heat loading. The charts give the full load plant performances at the reference atmospheric conditions of 49 F wet bulb and 52 F dry bulb, and the procedure illustrated on Table II may be used to find the performances the cooling towers would give under different atmospheric conditions.

A comparison of Figs. 3a and 3b shows that the economic water temperatures for mechanical draught towers are lower than those for natural draught towers over the

* In preparing this article the authors have made use of information and some of the diagrams contained in the papers cited in references (1) and (2), which were published in the Proceedings of the Institution of Electrical Engineers, Part A, Number 3, Volume 102, June 1955.
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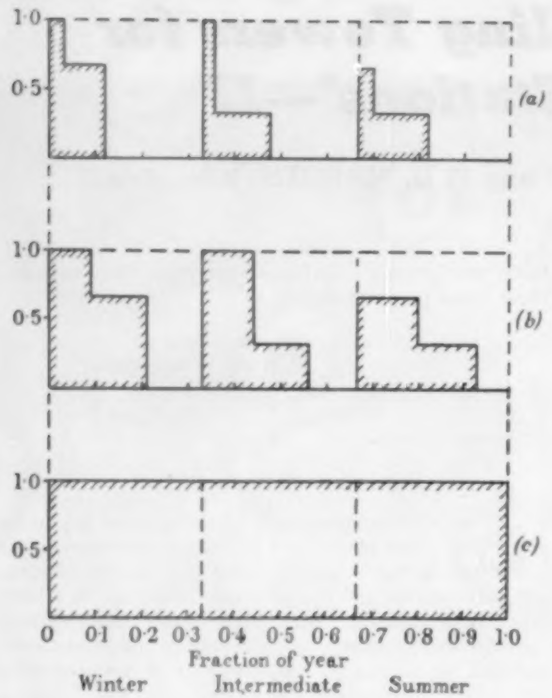
‡ Formerly with Kennedy & Donkin, London.
[1] Numbers in parentheses refer to Bibliography at the back of the article.



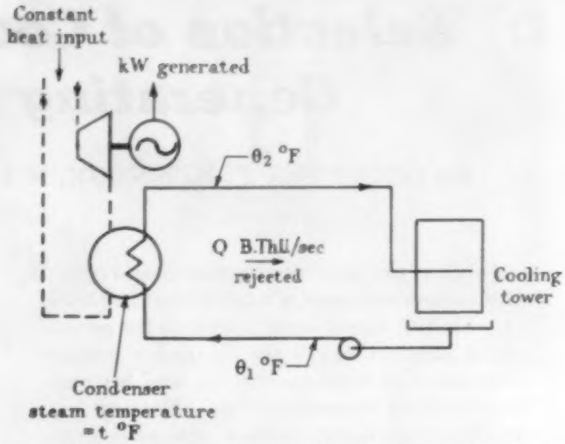


Fig. 2—The major components of the heat rejection circuit in a typical plant are the condenser, circulating water pump and the turbine exhaust. The duty of each affects the others

Fig. 1—Seasonal block load allocation diagrams employed in constructing Figs. 3, 4 and 6. Curve (a) is 20 per cent annual load factor; (b), 45 per cent annual load factor; and (c), 100 per cent annual load factor

relevant range of duty conditions, so that, when tenders for both types of plant are desired, different performance specifications should be formulated.

Fig. 4 indicates that the economic values of the water temperature ranges $(\theta_1 - \theta_2)$ are higher than those generally used in the past, probably because past practice was unduly influenced by the much lower economic water temperature ranges applicable for river-cooled generating stations.

With regard to the calculated economic temperature differences across the condenser, $(t - \theta_m)$, represented by the broken lines on Fig. 4, it is necessary to make the reservation that cost relations for condensers are more variable than those for cooling towers, particularly because the condenser dimensions may influence the dimensions and costs of the power station building. The condenser cost relations used for this article are however considered to be sufficiently representative to fulfill the main object, i.e., the representation of charts giving realistic assessments of economic cooling tower duty conditions.

Figs. 3 and 4 show that an increase in the turbine exhaust heat loading increases the economic values of all three temperature differences by very substantial amounts, and therefore reduces the economic physical dimensions of cooling towers, condensers and circulating water plant for a given heat load. This trend became apparent in Britain as turbine ratings increased from 30 to 60 and 100 mw with corresponding exhaust heat loadings for twin flow machines of about 1100, 1500 and 1900 Btu/sec-ft^2 respectively. As turbine ratings increase still further, even the adoption of multi-flow exhausts will not suffice to arrest the increase in the heat loadings of the turbine exhausts, and consequently cooling tower and condenser specifications involving still higher temperature differences will become economic.

Figs. 3 and 4 also indicate that an increase in load factor reduces the economic cooling tower and condenser temperature differences, but has little effect on the economic water temperature range, $(\theta_1 - \theta_2)$.

Fig. 5 illustrates the effect of climate on the economic duty conditions. To simplify the calculations for this chart it was assumed that constant atmosphere conditions would be maintained throughout the year, and that the towers would be on full load for 50 per cent of the year and on no loads for the remaining time. The figure clearly shows the reduction in the economic values of $(\theta_m - t_{w1})$ with increasing climatic temperatures.

Natural Versus Mechanical Draught

In comparing the economies of natural and mechanical draught towers, four cost items have to be taken into consideration: (*a*) The annual investments costs, (*b*) The annual cost of fan power, (*c*) The water temperatures attained under various operating conditions, and the effect on station costs, (*d*) The costs associated with the tower pumping head (which may differ for the two types).

At the economic design conditions, the investment costs for the two types are often comparable, with the mechanical draught tower having the advantage of attaining lower water temperatures, and the disadvantage of incurring fan power costs.

Fig. 6 shows the comparison of the total annual costs of the two types of tower, the area below the heavy shaded line representing the range of conditions over which mechanical draught towers are estimated to be financially superior to natural draught towers. The other curved lines on the chart show the net difference in

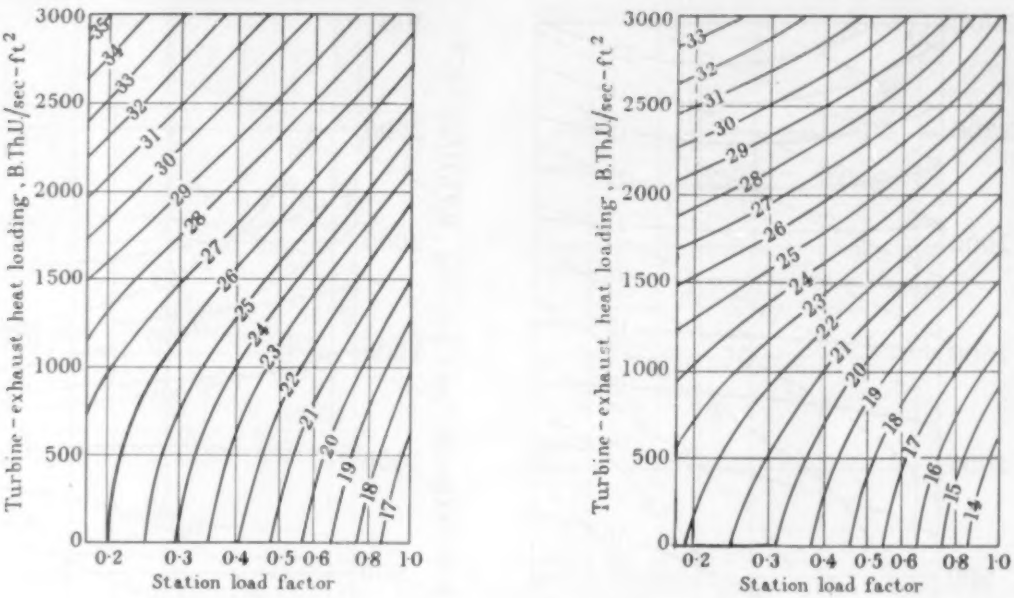


Fig. 3—Economic mean temperature difference of cooling towers (mechanical draft, curves above and natural draft, curves to the right) indicate what temperature difference ($\theta m - tw.$) is to be specified at the reference atmospheric conditions: wet bulb of 49 F and dry bulb of 82 F. Here economic water temperature is lower for mechanical draft

the annual costs expressed in £/annum per Btu/sec of the full load heat rejection. A difference of 0.01 for instance represents £1,000 per annum (i.e., about $2800 per annum) for a 60-mw set. The chart should not be used as a firm basis of selecting between natural and mechanical draught towers because site conditions do affect the relative prices to some extent, and makers' quotations are not always offered on the same basis. Nevertheless the chart gives a good indication of the main factors which influence the financial comparison under British price conditions.

Influence of Exhaust Heat Loading

Inspection of the chart indicates that the exhaust heat loading has a very strong bearing on the comparison, a high heat loading being in favor of natural draught. To explain this, one may regard a natural draught chimney as an engine which develops useful air power in proportion to its height and the amount of heat supplied to it. The heat is a waste product, so that the chimney is economic if the power developed from this free heat is sufficient to pay for the capital charges. Now Fig. 3b shows that an increase in the turbine exhaust heat loading produces an increase in the economic water temperatures, and hence a reduction in the economic ground area of the chimney. In other words, the chimney can still develop the same air power but costs less to construct, and is consequently more economic. Hence high turbine exhaust heat loadings are advantageous to natural draught.

Influence of Load Factor

Fig. 6 also shows that, contrary to common belief, station load factor has little influence on the economic comparison between natural and mechanical draught towers. The prevalent belief that natural draught towers tend to be more economic at high than at low load factors is probably based on the supposition that a chimney developing free air power continuously is in a better position to pay for its capital charges than one which develops power for only a small fraction of the year. That argument is correct, in itself, but there are other factors which cancel out its effect. The most important of these is the fact that with higher load factors larger towers (i.e., chimneys with larger ground areas) are economic as shown by Fig. 3b. Hence the cost of the chimney also increases. Furthermore, for a given heat quantity supplied to it, a cooling tower chimney can develop more power on cold days than on hot days. Low and medium load factor stations tend to spend more of their operating time in the winter than in the summer—and from that aspect are favorable to natural draught. Stations with very high load factors (i.e., approaching 100 per cent) must inevitably be loaded almost as heavily in the summer as in the winter. Fig. 6 shows that the various factors mentioned above approximately cancel, i.e., that station load factor has little influence on the economic comparison.

Influence of Climate

Fig. 7 shows the dividing lines between the economic spheres of application of mechanical and natural draught towers for various climatic conditions. Each curve is based on one wet bulb temperature and an associated dry bulb temperature which are assumed to be maintained throughout the year. Moreover, to simplify the calculations for this chart, it is assumed that the towers are never on partial load, so that a load factor

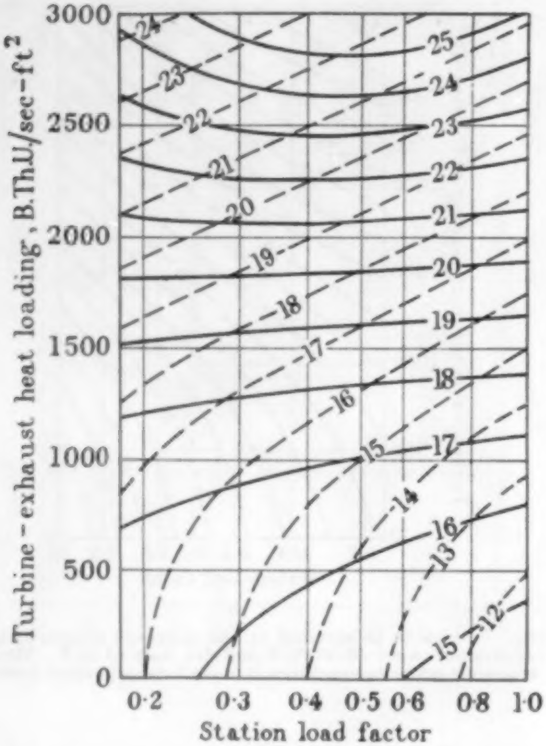


Fig. 4—Economic water temperature range and condenser temperature difference as defined in text under "Economic Duty Conditions" is shown for natural draft cooling towers but apply within ±1 degree F for mechanical draft cooling

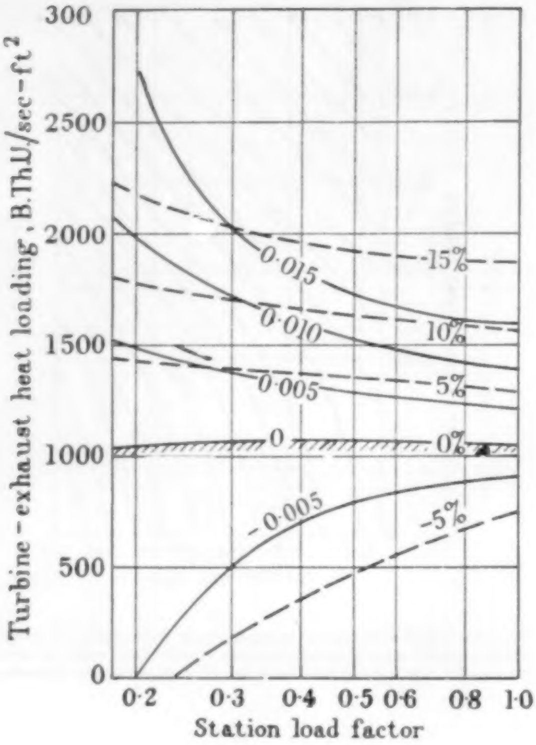


Fig. 6—The total costs of generation with mechanical draft towers less the total cost of generation with natural draft towers are shown in the solid line curves in terms of British pounds per year per Btu per second of full load heat rejection

of 40 per cent is, for instance, obtained by operating the tower on full load for 40 per cent of the year.

The figure shows the very pronounced influence of the climate. With a mean wet bulb temperature of 40 F there is no economic application at all for mechanical draught towers, however low the exhaust heat loading; for a mean wet bulb temperature above 70 F there is no economic application for natural draught towers, for any foreseeable turbine exhaust heat loading. Northern Canada and India are examples of these two extreme climatic conditions. The U. S. has average air temperatures about 10 F higher than Britain and this has no doubt contributed to the current predominance of mechanical draught towers in U. S. power stations, and the predominance of natural draught towers in British stations. The precise positions of the dividing line on Fig. 7, of course, depend on the cost data and investment rates, and modified data may apply in countries other than Britain.

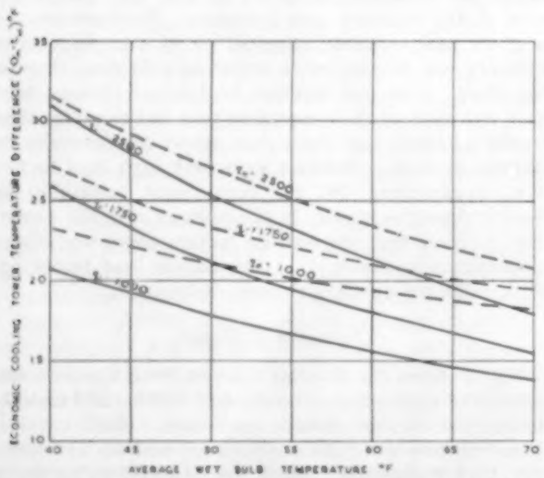


Fig. 5—Curves show the conditions under which both mechanical and natural draft cooling towers give the same overall cost of generation. Natural draft are economic above the curve. The dotted line is British conditions.

With the uniform load and temperature conditions assumed for each curve on Fig. 7, the station load factor does influence the economic comparison between natural and mechanical draught towers to a small extent.

Economic Operation

With natural draught towers fitted with water distribution systems which work effectively over a wide range of water quantities it does not pay to shut down some of the towers however light the station heat load. The only independent operating variable is thus the water quantity, and this should be reduced by cutting down the number of pumps on load whenever the incremental generator output reaches a low value, i.e., whenever a greater economy can be obtained by reducing the pumping power than by improving the vacuum. This

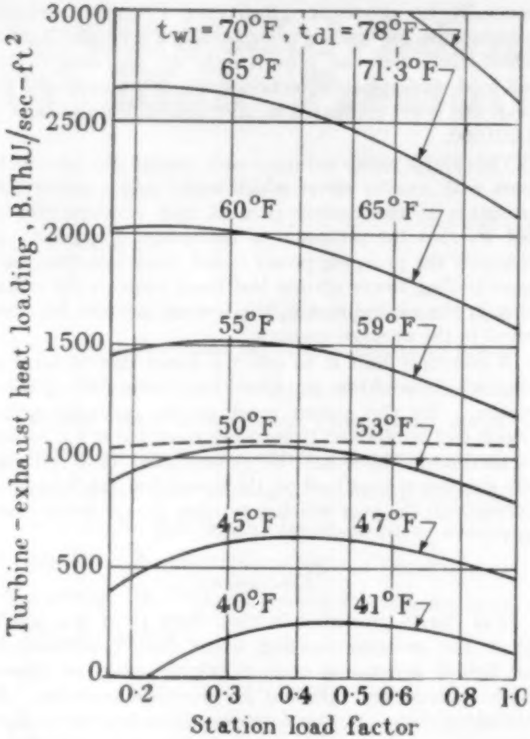


Fig. 7—Above curves picture the conditions under which natural and mechanical draft towers give the same overall costs of generation. The areas above the curve lines represent the range wherein natural draft towers are economic.

reduction in the number of pumps on load should take place particularly in the winter and at times of low station load.

With mechanical draught towers it does pay to cut out some of the cells at times of light station load or low atmospheric temperatures in order to reduce the fan power consumption.

Calculations for this article are based on single speed motors, but for certain ranges of condition the extra expense of two speed or even variable speed motors is justified by the reduction in the average fan power consumption. The net change in the overall annual costs is not, however, sufficient to change Figs. 6 and 7 appreciably.

Combined River and Cooling Tower Schemes

Where a direct choice exists between the use of a self-contained river cooling scheme and a cooling tower scheme, the former would almost invariably be chosen because the river works usually cost much less than the cooling towers, the average vacuum obtained by river cooling is better, and the pumping costs are lower. Altogether these factors may account for a capitalized saving of some £800,000 for a 240-mw station.

There are, however, few rivers in Britain large enough to deal with the entire heat rejection of modern generating stations during especially dry periods. More often, combined river and cooling tower schemes have to be adopted, and these can be arranged in two basic types of

TABLE II—EXAMPLE OF DESIGN* AND ASSESSMENT OF PERFORMANCE OF TOWER AT OTHER ATMOSPHERIC CONDITIONS

Item	Amount: Mechanical Draught	Amount: Natural Draught
1. Design		
(1) Given data:		
(a) Turbine exhaust heat loading	2000 Btu/sec-ft²	
(b) Station load factor	0.5	
(c) Climate	As shown on Table I	
(2) Economic ($\theta_m - t_{wl}$) based on reference atmospheric conditions, t_{wl} = 49 F, t_{dl} = 52 F: (from Fig. 3), deg F	25.2	26.6
(3) θ_m for reference conditions = (2) + 49 F, deg F	74.2	75.6
2. Performance of Above Tower on Hot Day (t_{wl} = 60 F, t_{dl} = 66 F)		
(4) ($h_m'' - h_l$) for reference conditions (from Fig. 3, Part I), Btu/lb	18.2	19.8
(5) Atmospheric factors, y_M & y_N: (from Fig. 4, Part I)		
(a) At reference conditions	1.420	10.23
(b) On hot day	1.451	11.50
(6) ($h_m'' - h_l$) for hot day [= (4) × (5b)/(5a)], Btu/lb	18.6	22.2
(7) ($\theta_m - t_{wl}$) for hot day (from Fig. 3), deg F	21.0	24.2
(8) θ_m for hot day, [= (7) + 60 F], deg F	81.0	84.2

* From Fig. 3.

NOTE: The variations in the atmospheric factors for mechanical draught towers are so small that it is usually permissible to omit steps 5 and 6 with these towers.

circuit, shown on Fig. 8. Circuit (a) has a shunt across the towers which enables the station to operate normally with river cooling, the tower being brought into operation only at times of low river discharge to prevent the river temperature from rising to unacceptable values. The provision of separate condenser pumps and cooling towers pumps minimizes pumping costs since most of the water is pumped only against the friction head in the condenser circuits, and not the static head of the tower circuit. Variable speed motors may be used for the cooling tower pumps to facilitate river temperature control. With fairly large rivers the tower would normally be used mainly in the summer, i.e. the season of low river discharges and high river temperatures, and in these circumstances mechanical draught towers would

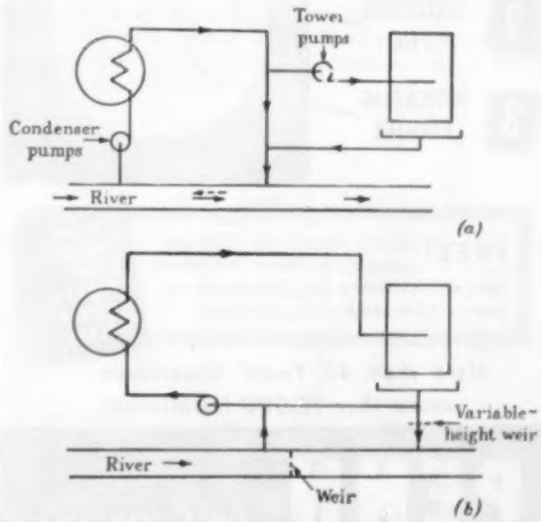


Fig. 8—Many installations employing river water for cooling find cooling towers can be added for times of low-river flow or high heat rejection. Circuit (a) uses a shunt across the towers and in this case the flow direction reverses in the river during very low river discharges. The other, circuit (b), is a simple series connection with a weir indicated although this can be eliminated

generally be economic. Reference (2) gives detailed calculations for such a scheme for a 240-mw station, where the river can absorb only 16 per cent of the full load station heat rejection in the driest period of the year, but is yet estimated to save a capitalized amount of £550,000.

The simple series arrangement, circuit (b), should be used with smaller rivers which make only a minor contribution to the cooling process, and consequently do not warrant the provision of two stages of pumps. In this case the pumping power is not much less than for a pure cooling tower station but there remains the reduction in the capital cost of the towers, and the improvement in the average vacuum.

A common fault is to select a tower just as large as that which would be employed for a pure cooling tower station. By this means much of the potential saving which can be derived from using river water for cooling is sacrificed. It should be remembered that it is not the maximum heat load on the tower, but the heat loads throughout the year which determine the economic tower size.

Conclusions

The charts presented in this, Part II of the article, show the economic cooling tower duty conditions for all British generating stations relying solely on towers, with an accuracy sufficient for practical purposes. By the use of these charts in formulating cooling tower specifications substantial savings can be made relative to designs based on past practice. The charts may be expected to apply also for the coldest districts in the U. S., and similar charts could be prepared for the warmer districts using the methods described in reference (2). Fig. 5 of this article gives an indication of the effect of climate.

Cooling tower duty conditions will become less onerous as larger turbine units with inevitably larger exhaust heat loadings come into general use. This trend of development will strengthen the financial advantage of the natural draught tower over mechanical draught units for British stations. It is only in combined river and tower schemes or in hotter climates that the mechanical draught tower becomes economic for generating station applications.

The use of local rivers for supplementary cooling should always be investigated, since large savings often result. With relatively large rivers a circuit should be adopted which avoids pumping all the water against the static head of the towers.

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Saving Molybdenum and Other Alloys in High Pressure Steam Equipment

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Availability of molybdenum and other alloying elements is critical in many areas of the world. Here is an interesting summation of a design philosophy advocated in some quarters to overcome or minimize the effects of limited alloying materials for high temperature steam service. In addition start-up recommendations differ considerably from prevalent U.S.A. practice.

A reduction in the consumption of molybdenum containing steels in superheaters can be effected by using molybdenum free steels for the section of the tubing where the temperature is below 450 C and using molybdenum containing perlitic steels only for the hotter section of the superheater. If tungsten alloyed perlitic steels prove satisfactory, the use of molybdenum free steels could be extended to temperatures up to 500 C. Another possibility is to replace molybdenum containing steels with austenitic steels, which can be used up to 600 C. From the cost point of view the latter solution is unfavorable.

To utilize fully the metal material, the temperature of the tubing should be as near as possible to that of the steam. For this purpose it is necessary to: (1) exclude any possibility of exceeding the positive tolerances of steam temperature at any spot of the superheater, (2) assure that the saturated steam fed into the superheater is free of any contamination so as to exclude any possibility of formation of salt deposits or of corrosion on the walls of the superheater tubes, (3) prevent salt contamination of the steam during the superheating stage (e.g., by the steam temperature regulation apparatus) and (4) ensure sufficient cooling of the superheater during starting up of the boiler. In this paper these enumerated factors are investigated and a design is described to fulfill these requirements.

Design of the Superheater

Disregarding the resistance to heat flow of the tube wall, the temperature difference of the tube wall temperature with respect to average steam temperature can be expressed by the formula:

$$\Delta t = \Delta t_p + \Delta t_R + \frac{\Delta t_{KP}}{\frac{\alpha r}{\alpha_K} + 1} \qquad (1)$$

where the symbols represent the following factors:

Δt = difference of tube wall to steam temperature

Δt_p = positive deviation from the average steam temperature along the width of the pass in the individual spirals of the superheater

Δt_R = admissible positive tolerance of the steam temperature regulation

Δt_{KP} = flue gas temperature change with respect to steam temperature drop

α_p = heat transfer coefficient, steam side

α_K = heat transfer coefficient, flue gas side

The temperature deviations, Δt_p, from the average steam temperature in the individual parallel sets of tubing along the pass width are due to non-uniformities in the temperature, speed and chemical composition of the flue gases, non-uniform thicknesses of ash deposits, non-uniformities in the specific gravity and viscosity of the steam in the individual tubes, and unequal distribution of the steam among the individual tubes.

Fig. 1A shows diagrammatically the characteristics of the steam temperatures and the flue gas temperatures of a counterflow type superheater assuming a 200 C difference between the temperatures of the right and the left sides of the pass for the inflowing flue gases. The steam has temperature of 360 C and is fed in from the steam temperature regulator. At the exit point from the superheater the difference in steam temperature between the left and the right sides is 84 C. If the mean steam temperature is 540 C, the positive deviation, Δt_p = 42 C, which is inadmissibly large.

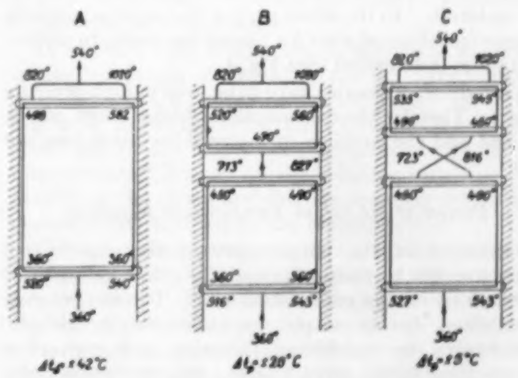


Fig. 1—Comparative steam temperatures and flue gas temperatures in parallel sets of superheater tubing under a straight counterflow design, A, a subdivided superheater with mixing of the steam from one stage to the next, B, and a system of crossing the steam paths in the superheater in passing from one stage to the next, C.

The value of Δt_v can be reduced by subdivision into two or several parts of the superheater and intensive intermediate mixing of the steam passed from one stage of the superheater to the next, Fig. 1B. A further very effective measure is to cross the steam paths in the superheater when passing from one stage to the next, Fig. 1C. Mixing and crossing bring about a pressure loss in the superheater.

In cyclone and slag tap furnaces the ash content of the flue gases is lower than in other types of furnaces and thus the danger of formation of non-uniform thicknesses of deposits on the flue side of the tubes is reduced.

From the point of view of the regulation of the steam temperature it is best to place the regulator between the first and second superheater stage. Thereby the response is quicker than if placed in front of the superheater and, for instance, in case of load increase only the mass of the second part of the superheater is cooled.

To ensure that the saturated steam is free of contaminations, high pressure boilers must be provided with a sufficiently large steam space and also the produced saturated steam has to be washed to reduce its content in dissolved salts.

To ensure low Δt_v and α_v/α_K values, the last stage of the superheater is placed into the rear pass and designed as a convection superheater with the flue gas and steam flows in the same direction. Thereby the temperature gradient and the heat transfer coefficient on the flue gas side is kept low. The heat transfer coefficient on the steam side depends only on the steam speed and steam parameters and the tube diameter. The disadvantage of this arrangement is that larger heating surfaces are required for the last stage, but this disadvantage is compensated by the higher reliability in operation.

For the highest steam temperatures austenitic steels are used in preference to perlitic ones, since their creep characteristics at about 600 C are more favorable. In this case joining of the austenitic to the perlitic section of the tubing is a major design problem. If a welded joint is made, it is very difficult to prevent deterioration of the material in the neighborhood of the weld joint due to diffusion of carbon and other alloying elements. At the high temperatures of the steam the diffusion can become very considerable. A further difficulty is the differing heat conductivity and thermal expansion coefficients of the two materials. In the latter part of this paper a specially designed mechanical joint for joining austenitic to perlitic steel tubes is described (See Fig. 4).

Welded-on accessories have to be used in high-pressure boilers. Thereby the consumption of alloy steels can be reduced and the problem of material for bolts does not arise.

Design of the Steam Temperature Regulator

Variations in the temperature of the superheated steam are due to fluctuations of the steam output and fluctuations on the combustion side. The temperature fluctuations due to output variations can be reduced considerably by combining radiation and convection superheating in the correct ratio. The fluctuations due to the processes on the combustion side cannot be easily controlled.

The fuel position in most countries is such that fuels of a great variety of characteristics have to be burned in boilers and even if the superheater characteristic is flat equipment capable of effecting rapid changes of steam temperature, as much as 150 C, must be used. High capacity steam temperature regulators are particularly important in slag tap furnaces where changes in the ash properties even of one type of coal bring about intensive fluctuations of the steam temperature.

A good steam regulator has to fulfill the following requirements: The maximum positive deviation of the steam temperature from the rated value should be as small as possible, should be of the shortest possible duration, the regulation should be aperiodic and the degree of regulation non-uniformities must be zero. Thus regulation with back coupling cannot be applied unless an elastic back coupling, or an additional impulse controlled by the direction and the speed of the temperature change is employed.

In modern boilers the steam temperature is controlled by basic and supplementary temperature regulation. The basic one operates for slow changes and is of the surface type or is controlled by such means as recirculation of the flue gases. The supplementary regulation is almost invariably of the injection type and intended to operate mainly when sudden changes occur in the steam temperature.

Superheater Protection During Starting-Up

The safest way to start up a boiler is by heating up with steam from another source before ignition of the fire. Thereby the danger of burning through of the superheater or the boiler tubes is excluded. For this purpose full pressure steam from the main or medium pressure steam can be used. The water in the boiler thus warms up slowly and comes to a boil. By further feeding of steam the pressure in the boiler increases to about 3 to 5 atmospheres. On reaching the required pressure, flow of steam into the superheater and thus also its cooling is ensured by venting the exhaust side. Then the fire can be started. Since a water-steam mixture is already present in the ascending tubes, there will already be a normal circulation in these. The heat generated in the combustion space will produce immediately further steam and thus the transition from heating up by outside steam to normal operation will proceed smoothly.

If outside steam is not available, it is advantageous to use a small electrically operated boiler for producing the necessary steam for starting up or to heat up the boiler water electrically instead of by outside steam. Such starting up also saves feedwater which otherwise would have to be used for protecting the economizer during the warming up period.

Equipment Design by the Vitkovice Steelworks

The high-pressure steam generator design of the Vitkovice Steelworks aims at solving all the above discussed problems taking into consideration the described difficulties and experience gained in practical operation.

The superheaters in the Vitkovice design are of the multistage type with mixing and crossing of the steam streams. The hottest part of the superheater is placed into the second pass of the boiler and is built as a unidirectional flow, convection type superheater. Fig. 2 shows a diagrammatic sketch of a steam superheater for producing 105 atmosphere, 550 C, steam..

After cleaning by the condensate the saturated steam

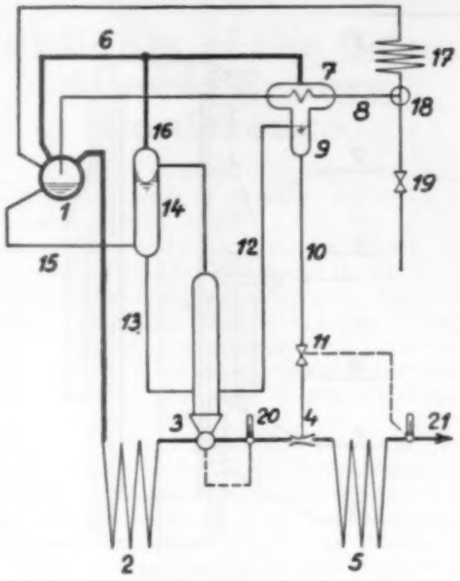


Fig. 2—Schematic sketch of a double superheated temperature regulation method—a combined surface condenser, 3, and an injector, 4,—to give control over final steam temperature as devised by the Vitkovice Steelworks, Czechoslovakia

is fed from the drum, 1, to the first superheater stage, 2. Following this superheater stage, the surface-, 3, and the injection-, 4, steam temperature regulators are arranged in series. After cooling the steam in the regulators, the steam is heated to the rated temperature in the second superheater stage, 5. Cooling of the steam in the regulators is effected by means of high-pressure condensate produced from saturated steam directly in the boiler.

For producing the high-pressure condensate washed saturated steam is drawn from the drum, 1, and fed into the surface condenser, 7, through the tubing, 6. There the saturated steam condenses at the full pressure under the cooling influence of the colder feedwater which flows through the tubing, 8. The condensate produced accumulates in the reservoir, 9, and from there it is fed through the tubing, 10, to the injection valve, 11, which controls the condensate injection into the steam in the regulator, 4. The high-pressure condensate produced from the saturated steam is of the same quality as the superheated steam and therefore injection of this condensate does not deteriorate the quality of the superheated steam.

The condensate in the reservoir, 9, has the same pressure as the saturated steam in the boiler drum, 1, and the injection of the condensate into the steam in the regulator, 4, is effected by the pressure difference resulting from the pressure loss in the first superheater stage plus the weight of the condensate column above the injection valve with a possible additional suction effect of the regulator if it is of the ejector type. Therefore no injection pump is needed.

The rate of feedwater supply to the boiler varies and therefore the reservoir, 9, covers the instantaneous differences between the consumption and production of condensate. Thus, the performance of the regulators, 3, and 4, is not dependent on the instantaneous rate of feedwater supply to the boiler. The amount of condensate produced is such that the consumption of the surface and injection regulators is covered with an adequate reserve of condensate.

The condensate which is not used by the injection regulator flows from the reservoir, 9, through the overflow tube, 12, into the surface temperature regulator, 3, where it becomes mixed with the recirculating water from the return tube, 13. The heat drawn off by cooling the steam in the regulator, 3, evaporates a part of the mixture and the water-steam mixture is transported through natural circulation into the cyclone separator, 14, in which the waters is separated from the steam. Then the water flows through the tube, 13, back to the regulator, 3. Since the amount of condensate flowing from the reservoir, 9, into the regulator, 3, is larger than the amount of evaporated condensate, the excess is made to flow from the separator through the tubing, 15, back to the washing drum of the boiler. Thereby the cooler-regulator, 3, is continuously flushed by condensate, preventing formation of undesirable concentrations. The intensification of undesirable concentrations due to evaporation in the regulator, 3, will become less the larger the amount of the excess condensate which flows back into the washing drum.

The steam separated out in the separator, 14, is fed through the tubing, 16, back to the tubing, 6, and thereby back into the condenser, 7. Since the flow of the excess condensate through the regulator, 3, prevents intensification of its solids content, the steam produced by evaporation in this regulator will be practically salt free. In fact, its quality will be better than that of the saturated steam produced in the boiler. Feeding of a practically salt free high-pressure condensate into the surface regulator, 3, prevents formation of solid deposits and thus also corrosion. Leakage in the regulator, 3, cannot bring about contamination of the superheated steam, a situation which would require an immediate shutdown of the boiler.

The boiler is provided with a sampling coil, 17, in front of which there is a three-way slide valve, 18, to distribute the water fed through the feed valve, 19, partly to the sampling coil and partly to the surface condenser, 7. The slide valve, 18, is set once only, namely during the test run, and it is set to supply sufficient cooling water even when burning coal which produces the highest temperature superheated steam.

In addition to other advantages, the described arrangement consists only of vessels and tubing. The number of moving parts are limited to the 3-way slide valve of the regulator, 3, and the valve of the injection regulator, 11. The regulators, 3 and 4 are so controlled that the surface regulator, 3, shuts off the injection regulator, 4, by precooling the steam and thereby reducing the quantity of condensate injected by the regulator, 4.

The steam temperature is measured by the control thermometer, 20, and maintained constant by the surface regulator. The constant temperature to which this thermometer is set is so chosen that the rated steam temperature is attained after the second stage of the superheater even at reduced boiler output. This temperature is practically the same as that of the steam temperature after the first stage of the superheater, 2, under the lowest boiler output for which the rated steam temperature can still be obtained no matter what fuel is burned which is a desirable design point.

The injection regulator, 4, is controlled by the thermometer, 21, and cools down the steam to such a temperature that it becomes heated to the rated temperature in the second stage of the preheater.

At low boiler loads it is not necessary at all to cool the steam and the entire condensate produced by the condenser flows through the surface regulator directly into the washing drum of the boiler.

A special safety valve is fitted behind the superheater for protection of the superheater in case of failure of the regulators, 3 and 4. This valve opens when the steam temperature increases above the permitted tolerance limit. By opening this valve the steam flowing through the superheater is instantaneously increased by the amount blown off by the safety valve, which leads to a decrease of the temperature to within the permitted limits and a shutting off of the safety valve.

The excess condensate flowing through the tubing, 15, into the drum, 1, is used for washing the saturated steam. The contact of the steam with the condensate on as large a surface as possible brings about the removal from the steam of a part of the salts dissolved therein, mainly silicate salts. A diagrammatic drawing of the steam washing equipment is shown in Fig. 3. The boiler is fitted with two series connected drums. The water-steam mixture from the riser tubes, 6, is introduced into the steam space, 7, of the drum, 1, where the water separates from the steam and flows back into the boiler through the tubes, 8, by force of gravity. The saturated steam is fed through the steam link tubes, 9, into the washing drum, 2. From the space behind the steel sheet, 3, of this drum the steam is led by the tubes, 4, below the perforated sheet, 5. The water space, 2, of this drum contains a mixture of feed water and of the excess condensate returned to the washing drum from the steam temperature regulation circuit.

The perforated sheet, 5, breaks up the steam flow into narrow streams from which the steam bubbles rise to the water surface. By passing through the water layer, the steam bubbles are purified from salt content. The thus purified steam flows into the superheater. The continuously fed-in feedwater and excess condensate flow into the washing drum, 2, and from there through the water link tubes, 10, into the main drum, 1.

In superheaters made partly of austenitic steel, the Vitkovice Steelworks use for the joint between the austenitic and the perlitic sections a mechanical joint the

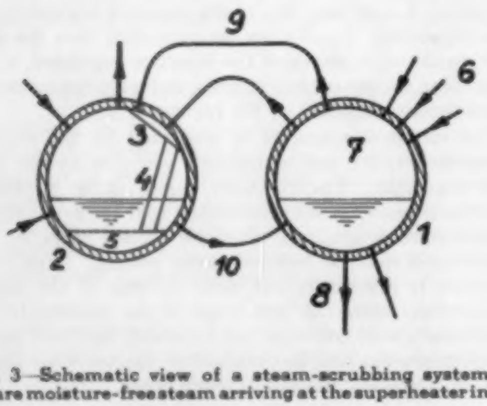


Fig. 3—Schematic view of a steam-scrubbing system to assure moisture-free steam arriving at the superheater inlet.

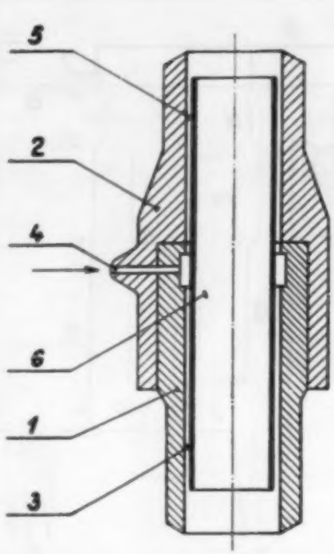


Fig. 4—Diagrammatic representation of a joint between austenitic and perlite metals in a single superheater tube to get around a shortage of alloy steels.

design of which is shown in Fig. 4. This joint employs a pressed fit, cooled by means of saturated steam to below 400 C to combat any loosening of the joint from creep phenomena. In addition, this cooling completely prevents diffusion between the austenitic and the perlitic steels. The amount of the saturated cooling steam is about 1 to 2 per cent of the superheated steam flowing through the tubing. The saturated steam flows into the joint with an excess pressure equal to the pressure loss in the perlitic part of the superheater.

As is shown in Fig. 4, the perlitic part, 2, which has the lower coefficient of heat expansion, is pressed onto the austenitic part, 1. A tube, 3, is inserted, at the inside, to carry the superheated steam past the joint. The space, 5, between the carrier tube, 3, and the outer joint, is filled with saturated steam, fed in by the tube, 4. The saturated steam cools the joint, and also forms an insulating layer inside the space, 5, insulating the joint from the hot carrier tube, 3. The length of this carrier tube must be large enough to prevent heating the joint by convection from parts at full steam temperature.

Highlights of the Geneva Atomic Energy Conference

At the United Nations Conference on The Peaceful Uses of Atomic Energy held in Geneva, Switzerland, from August 8–20, more than 400 scientific and technical papers were presented by representatives from 72 nations throughout the world. Subjects of these papers ranged from reports on fundamental properties of physical particles through topics in nuclear chemistry, medicine, geology, and applied engineering. In this brief report it is possible to highlight only a few of the many excellent papers that pertain to the field of nuclear power generation.

In a paper entitled "Economics of Nuclear Power" **James A. Lane** of the Oak Ridge National Laboratory outlined the AEC five-year Civilian Power Reactor Program and the Industrial Power Demonstration Reactor proposals thus far placed before the Commission.

The AEC prototype reactor program and the industrial full-scale proposals both deal with the types of reactors which studies have shown offer the most promise for competitive power. These are: *a*, the pressurized water reactor (PWR); *b*, the boiling reactor (BWR); *c*, the sodium graphite reactor (SGR); *d*, the fast breeder reactor (FBR); and *e*, the aqueous homogeneous thorium breeder reactor (TBR). Schematic diagrams of these reactors are shown in Figs. 1 through 5. In the *pressurized water reactor* (Fig. 1), a full-scale version of which is now being constructed by Westinghouse, either heavy water or ordinary water can be used as the coolant and moderator with appropriate adjustment of the spacing and enrichment of fuel elements. (The Westinghouse reactor uses H_2O.) The water is pumped between the solid fuel rods in the reactor core and then to a heat exchanger where steam is produced to drive a turbine. In the *boiling water reactor* (Fig. 2) (also using either H_2O or D_2O), steam is produced directly by allowing boiling to occur within the reactor core. This method of heat removal eliminates the heat exchanger, has corresponding higher thermal efficiencies and reduces the size of pumps and other equipment. Offsetting these advantages are the required higher fuel enrichment due to the presence of steam in the core and the necessity for enclosing the turbine and condenser within a shield due to the radioactivity of the water.

The *sodium graphite* reactor (Fig. 3) takes advantage of the high temperatures and high thermal efficiencies to be gained through the use of liquid sodium as the coolant circulates between solid fuel elements in the reactor and then through a heat exchanger where heat is transferred to a secondary sodium system. In contrast to the primary sodium, which becomes radioactive, the secondary sodium can be circulated outside of the shield through a steam generator. The use of sodium as a high temperature coolant is also applicable in the case of a *fast breeder reactor* (Fig. 4) Here the reactor consists of an unmoderated core, fueled with plutonium containing some uranium-238, and surrounded by a uranium-238 blanket. Primary sodium is used to cool both the core and blanket in conjunction with a secondary sodium system, as in the case of the SGR. The advantage of the fast breeder is the high breeding gain

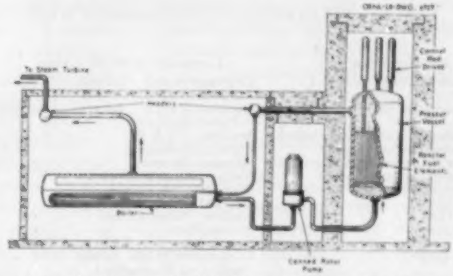


Fig. 1—Pressurized Water Reactor System (Forced Circulation H_2O or D_2O)

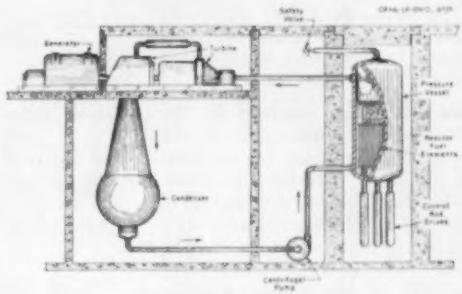


Fig. 2—Boiling Water Reactor System

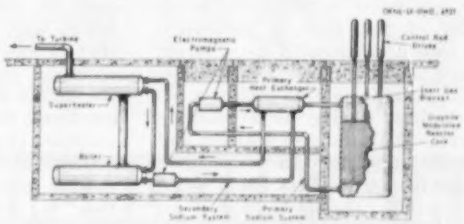


Fig. 3—Sodium Graphite Reactor System

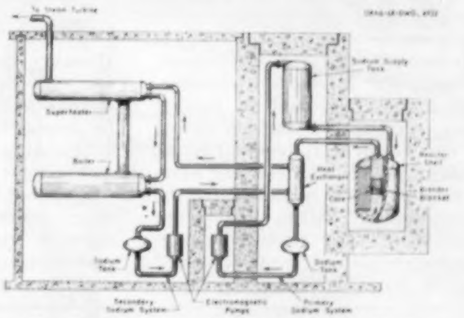


Fig. 4—Sodium Cooled Fast Reactor System

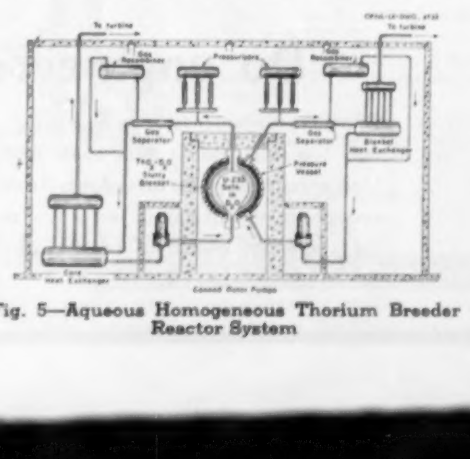


Fig. 5—Aqueous Homogeneous Thorium Breeder Reactor System

TABLE I—ESTIMATED CAPITAL COSTS OF NUCLEAR POWER PLANTS

Reactor Type	Company or Study Group	Power Level Mw-electricity	Capital[a] Cost, $/kw
D_2O moderated and cooled	Commonwealth Edison (Nuclear Power Group)	234	352
H_2O moderated and cooled	Commonwealth Edison (Nuclear Power Group)	273	236
D_2O moderated and H_2O cooled	Bechtel, PG & E	101	365
H_2O moderated and cooled	Fluor Corporation	200	183
Graphite moderated, H_2O cooled	General Electric	700	249
H_2O moderated and cooled	Yankee Atomic Electric	100	240[b]
H_2O moderated	Consolidated Edison	236	233[b]
Boiling H_2O	General Electric	300	226
Boiling H_2O	Nuclear Power Group	180	250[b]
Sodium graphite	Monsanto	150	291
Sodium graphite	North American Aviation	150	243
Sodium graphite	Consumers Public Power District of Nebraska	75	323[b]
Fast breeder	California Research and Development	173	269
Fast breeder	Detroit Edison	100	450[b]
Aqueous homogeneous	Foster Wheeler-Pioneer Service	100	256
Aqueous homogeneous	Nuclear Power Group	180	240

[a] Costs do not include fuel inventories in reactor and processing plants.
[b] Power Demonstration Reactors—1955.

possible with fast neutrons in the Pu-239, U-238 system, due to the lower parasitic capture of neutrons by Pu-239 itself. On the other hand, fast reactors require a large amount of nuclear fuel to achieve criticality with corresponding high inventory charges.

Finally, the *aqueous homogeneous reactor* (Fig. 5) uses a dilute solution of U-233 in D_2O flowing through an inner core vessel, in which the major portion of the heat is generated. This core is surrounded by a larger pressure vessel through which a suspension of ThO_2 in D_2O is circulated, the thorium serving as a breeding blanket. Both the core solution and blanket suspension are circulated through heat exchangers where steam is generated. The advantage of using a fluid fuel and blanket is primarily that of having simpler chemical processing systems and thus lower fuel processing costs. Both the uranium solutions and thorium suspensions, however, are somewhat corrosive, highly radioactive, and in general require very special fuel-circulating and heat-removal systems. One of the major problems in handling aqueous fuel solutions is caused by the decomposition of D_2O in the presence of radiation. The explosive deuterium-oxygen gas mixture thus formed must be separated from the circulating liquid stream, as shown in Fig. 5, and recombined before being injected back into the system. The pressurizers shown serve to maintain a constant volume and prevent boiling in the circulating liquid.

Capital Costs

From Table I it is possible to get a broad view of the range of capital costs for nuclear plants now in the design stage. Mr. Lane pointed out that in most cases, estimated power costs are more strongly affected by the various non-technical assumptions employed, such as plant write-off, inventory charges and load factor, rather than by factors related to reactor technology. Since the selection of non-technical assumptions reflects the degree of optimism or pessimism of those making the estimate, one can say that such projected costs have no real significance. However, the fact that the optimists outweigh the pessimists and most predictions indicate a promising outlook for competitive nuclear power is very encouraging. As a matter of fact, most people are asking the question, "*When* will we have competitive nuclear power?" Estimated capital costs of various large-scale nuclear power plants summarized in Table I vary from $183 per kw capability to $450 per kw. Corresponding power cost estimates range from 4 mills per kwhr to 10 mills per kwhr.

Choice of Reactors

Alvin M. Weinberg of Oak Ridge National Laboratory in a paper entitled "Survey of Fuel Cycles and Reactor Types" took up the question of which types of reactors might emerge as the most widely used. He noted the historical reality that there has been a tremendous development and corresponding high rate of obsolescence of power plant equipment. He argued that low thermal efficiency, which is the main reason for conventional thermal power generating equipment becoming obsolete, will not likely function in the same manner for nuclear power plants. Instead he expressed the belief that nuclear plants should be considered more analogous to hydro-electric plants: that is if they have sufficiently low overall operating costs, it is not obvious why they should become obsolete any more than dams become obsolete. It follows that in choosing lines of development the aims will always be to reduce operating costs. On this account, Dr. Weinberg continued, it is likely that ingenuity in fuel handling may prove more significant than ingenuity in the method of heat removal, for the former aims at low operating costs while the latter is concerned with low capital costs. This is admittedly a long run viewpoint, but it is one that will have some effect upon ultimate reactor choice.

Homogeneous Power Reactors

Advantages and disadvantages of aqueous homogeneous power reactors were detailed by **R. B. Briggs** and **J. A. Swartout** of Oak Ridge National Laboratory. Among the advantages derived from this type of reactor are the following:

1. Aqueous homogeneous reactors can be designed for high-power density and low fuel inventory. The critical mass of fuel is low, and the

rate that heat can be removed from a reactor may be limited only by the rate that the fuel can be circulated and the temperature rise which is permitted. Heat is removed from the fuel in equipment which is external to the reactor.

2. Reactor fuels can be purified continuously to remove fission products and radiation damage products. Fuel preparation, handling, and reprocessing are simplified and high burn-up of fuel is achieved.

3. The reactors possess a very high degree of nuclear stability. Gross changes in operating conditions can be achieved by adjusting the fuel concentration. The strong negative power coefficient provides regulation and prevents harmful power excursions. Control rods or plates are unnecessary.

4. Simple mechanical design results from the use of a fluid fuel and moderator and the elimination of control elements. The reactor vessel can be a tank containing only those parts required to achieve the desired flow distribution and, in some designs, to separate the reactor into regions containing fissionable or fertile materials.

5. Absence of structural materials in the reactor core, use of heavy water as the moderator and continuous removal of fissionable product poisons make it possible to minimize the parasitic absorption of neutrons.

The use of fluid fuels and an aqueous moderator also results in undesirable characteristics which include:

1. High pressures are associated with the high temperatures which are required for producing power efficiently. The vapor pressure of water is 1200 psig at 572 F and 2400 psig at 662 F. Circulating fuel reactors are operated with added pressure to prevent cavitation in the pumps and to minimize the volume of gases produced by decomposition of the moderator.

2. Large volumes of very highly radioactive materials are present in a fluid state at high temperature and high pressure. Provision must be made for containing those materials if they leak from the reactor vessels or piping. Pumps, piping and heat exchangers become radioactive from contact with the fuel. Their replacement or repair can be difficult and expensive.

3. Uranium, plutonium and thorium compounds, water and materials of construction form complex physical and chemical systems. Limitations on operating conditions are imposed by physical and chemical characteristics of the fuels and by corrosion and erosion effects.

4. Deuterium and oxygen are produced by radiolytic and decomposition of the moderator. They must be recombined safely.

Aqueous homogeneous reactors rely for control completely on variable fuel concentration and temperature coefficient of reactivity. Shim control to put the reactor into operation, to change the operating temperature and to compensate for fission product poisons is accomplished by adjusting the core fuel concentration. Dependence is placed upon the negative temperature coefficient of reactivity to meet the regulation and safety requirements of the reactor.

Reactor Startup

The reactor will be started by an orderly charging of the fuel into the core and blanket systems. The slurry will be pumped directly from the storage tanks into the blanket circulating system. At the same time, heavy water will be evaporated from the core solution in the tanks and pumped into the core circulating system. When both the core and blanket systems are filled, they will be pressurized and the circulation

will be started. At this point it will be desirable, although not absolutely necessary, to heat the circulating liquids to about 400 F by injecting steam into the heat exchangers. Then concentrated fuel will be added at a controlled rate. When the critical concentration is reached, the chain reaction will begin and the reactor temperature will rise with continued addition of fuel. When the temperature approaches the specified operating temperature, steam will be extracted from the heat exchangers to start the turbine. Final adjustments will be made to the concentration with the turbo-generator plant operating at full output. Excess fluid resulting from the expansion of the liquid in the reactor will be discharged to the storage tanks.

The process will be reversed in a normal shutdown of the reactor. Fuel from the core will be discharged to the dump tanks where the heavy water will be distilled and returned to the reactor. In this way the operating temperature can be reduced slowly until the reactor is cool. Depending upon the reason for the shutdown the reactor may be started again by injecting the concentrated fuel that will have accumulated in the storage tanks. In an emergency the reactor fluids can be discharged rapidly into the storage tanks.

Fast Reactors

Four representatives of the Argonne National Laboratory, **A. H. Barnes, L. J. Koch, H. O. Monson,** and **F. A. Smith**, presented a paper entitled "The Engineering Design of EBR-II, A Prototype Fast Neutron Reactor Power Plant." This plant is divided into four major systems which may be defined as follows:

1. The primary system: the reactor and the primary sodium cooling system.
2. The secondary system: the intermediate sodium heat transfer system.
3. The steam system: the steam-electric system.
4. The fuel process system: the fuel recovery and fabrication facilities.

The primary system is contained in a single vessel (referred to as the "primary tank"). All of the components in the primary system, including the reactor, the primary sodium pumps and piping, the heat exchanger, and the fuel transfer and storage system, are submerged in sodium.

The reactor consists of an enriched core of uranium-plutonium alloy, in the approximate shape of a hollow cylinder, surrounded on all sides by a uranium breeding blanket. The

average core power density is approximately 1000 kw/liter, and the average core heat flux is approximately 1×10^6 Btu per sq ft-hr. Reactor cooling is a critical problem, not only during operation, but also after shutdown. Considerable attention, therefore, is given to the primary cooling system.

The fuel and blanket material are contained in subassemblies of hexagonal cross section positioned vertically in a close-packed array. The coolant enters the bottom of each subassembly, flows upward through the subassembly, and then passes from the reactor through the intermediate heat exchanger.

The reactor loading and unloading process is carried out with the subassemblies completely submerged in sodium. Shutdown cooling requirements of the fuel dictate this unloading procedure. The subassemblies are transferred to the submerged storage racks, where the fuel continues to cool by natural convection of the sodium.

The heat is transferred from the heat exchanger to the steam generator by the secondary sodium system. This system is non-radioactive and serves to isolate the radioactive sodium in the primary system from the steam generator, and also to isolate the reactor from the moderating effect of water. Conventional piping and pumping arrangements are employed in this system, which is accessible for normal maintenance.

The steam generator is a sodium-to water, steam heat exchanger. Steam at 850 F and 1250 psig is supplied to a conventional turbine-generator of 20,000-kw capability.

The plant includes an integral fuel and fabrication facility to determine the feasibility and cost of a fuel cycle specifically designed to meet the needs of a fast power reactor. Low reprocessing costs and a short cooling time, to minimize fuel inventory, are desired. A direct metallurgical separations process has been selected for EBR-II fuel recovery which promises to accomplish both of these objectives.

Sodium Graphite Reactor

In a paper entitled "Sodium Graphite Reactor 75,000 Electrical Kilowatt Power Plant," **Chauncey Starr,** vice president of North American Aviation, Inc., and nine of his coworkers pointed out some of the important features and advantages of this system of nuclear power generation. These include the following:

1. A high coolant temperature (925 F at present, 1200 F projected) is possible without requiring pressurization of the reactor heat extraction system. This permits good steam

conditions and high thermal efficiency in the power conversion equipment.

2. The reactor can be adapted to a variety of fuel elements and operating conditions. Both slightly enriched uranium or a Th-U235 fuel may be used.

3. There are no chemical incompatibilities in the fuel element, coolant, structure combination.

4. The absence of releasable potential energy from pressurization or chemical reactions increases the inherent safety of this reactor system and permits the plant configuration to be easily arranged to contain radioactivity under all circumstances.

In selecting the steam cycle for a nuclear steam-electric station a compromise must be made between (1) maximum heat generation from a given reactor and (2) maximum conversion of this heat into electric energy, i.e., maximum thermal efficiency.

Heat removal from a reactor fuel element depends on the temperature drive from the center to the surface. Due to the excellent heat transfer characteristics of sodium, the temperature gradient between the fuel element surface and the sodium coolant is quite small. Therefore, heat removal from the reactor depends primarily upon the difference between the center temperature of each fuel element and the average temperature of the liquid sodium flowing up and through each fuel channel. The present state of metallurgical development sets an upper limit upon fuel element center temperature. Therefore, an increase in heat generation, at constant sodium velocity can only be accomplished by lowering the average sodium temperature.

The conversion of thermal power to electrical power (i.e., thermal efficiency) decreases with a decrease in the average sodium temperature. For a given heat sink, the efficiency of any heat cycle is determined by the average temperature of heat addition to the system. Thus, in order to obtain a high thermal efficiency, a fairly high average sodium temperature is required.

The non-reheat regenerative steam cycle appears to be the optimum choice at this time. In conventional fuel fired stations fuel economy can be considerably increased by the use of five or more extraction feedwater heaters, which raise the temperature of the feedwater to about 400 F. In a nuclear plant, however, a high feedwater temperature has a detrimental effect. It either lowers the maximum possible steam pressure with a consequent loss in cycle effi-

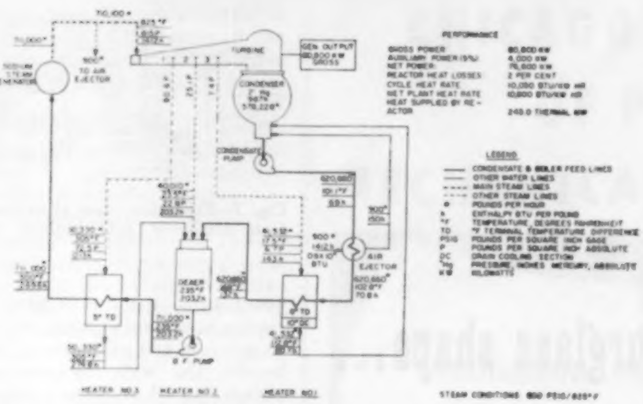


Fig. 6—Preliminary heat balance for 75-mw central station sodium graphite reactor

ciency, or it decreases reactor power output. The decrease in reactor power output does not result in any significant reduction in either capital expenditures or operating costs, whereas the higher feedwater temperature is only obtained by the installation of five or more heaters at a cost of about $50,000 per heater. Previous studies have shown that the best regenerative feedwater temperature for nuclear plants is in the range from 250 F to 300 F.

In the design under consideration the mean temperature of the primary sodium leaving the reactor has been limited to 925 F and a non-reheat regenerative cycle with 800 psig, 825 F throttle conditions and a feedwater temperature of 300 F has been chosen. The choice of 800 psig, 825 F represents what appears to be the best compromise between a low cycle heat rate, adequate temperature drives in the sodium-to-water heat exchangers, and permissible moisture in the turbine exhaust.

A three-stage feedwater-heating arrangement was selected to obtain a final feedwater temperature of 300 F. Preliminary heat balance calculations show a turbine cycle heat rate of 10,050 Btu per kwhr, as shown in Fig. 6.

Heavy Water Reactors

In a paper entitled "Heavy Water Reactors for Industrial Power, Including Boiling Reactors" **H. P. Iskenderian, L. E. Link, M. Treshow,** and **J. M. West** of Argonne National Laboratory stated that heavy water comes very close to meeting technical specifications for an ideal moderator material. Of the materials which are commonly considered as moderators for reactors, the slowing down power of heavy water is second only to light water. The combination of good moderating power and low absorption cross section for thermal neutrons gives heavy water a moderating ratio (slowing down power divided by absorption cross section) far greater than any other material.

In the past the chief disadvantage of D_2O as a moderator has been its high cost which ranged up to $100 per lb. Considering the amount of D_2O required per megawatt of heat output, the cost of the D_2O alone amounted to between 1.4 and 7 mills per kwhr of saleable electricity. More recent quotations for D_2O in the United States have shown a reduction in price to $28 per lb. At this price and with reactor designs which minimize the D_2O investment per unit of heat output, the direct charges against D_2O can be reduced to less than 1 mill per kwhr of electricity produced. This charge for D_2O then becomes significantly less than the cost of the fuel in a heavy water reactor and very much less than the cost of fuel in reactors using other moderators in conjunction with enriched uranium.

Direct-cycle boiling reactor power plants are particularly well suited to use of D_2O as moderator and coolant. The only D_2O required outside the reactor is a very small amount in the form of steam and condensate. In a pressurized water type the amount of D_2O required in external pipes, boilers and pumps approximates that in the reactor itself.

The advantages of D_2O moderator are best realized in reactors of large physical size. To achieve criticality and operate at high temperature with natural uranium fuel, D_2O-moderated reactor cores must be 10 ft or more in diameter. When the reflector is taken into account, a pressure vessel having a diameter of at least 13 ft is required. For a boiling reactor generating 600 psi steam, the thickness of the pressure vessel wall would be

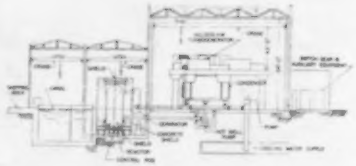


Fig. 7—Elevation view of central station 1000-mw D_2O boiling reactor

about 4 in. For a pressurized water reactor operating at 2000 psi in order to generate 600 psi steam in external boilers, the wall thickness of the pressure vessel would be so great that fabrication would be extremely difficult, if not impossible.

The choice of an optimum reactor design for power production depends upon the assumptions which must be made regarding availability of slightly enriched uranium and on the amount of power desired per reactor. If only natural uranium is available, D_2O reactors are necessarily large, and H_2O reactors are excluded entirely. A D_2O natural uranium reactor, which is large in physical size and capital investment, must produce large quantities of heat if the unit cost of this heat is to be reasonable. The design of a central station with a boiling D_2O reactor, utilizing natural uranium fuel, capable of producing 1000 megawatts of heat is shown in Fig. 7.

Russian Atomic Power Station

In a paper entitled "The First Atomic Power Station of the USSR and the Prospects of Atomic Power Development," **D. I. Blokhintsev** and **N. A. Nikolayev** of the USSR stated that the first industrial atomic power station in that country went into service on June 27, 1954, generating electricity from the energy of fission of uranium nuclei. The authors described some of the studies and experiments that were carried out by the Academy of Sciences of the USSR and provided a bibliography showing eighteen papers relating to the development of nuclear power plants.

Designed for a power output of 5000 kw, the atomic power station contains a pressurized water-cooled thermal uranium-graphite reactor with a rated heat-generating capacity of 30,000 kw. The average flux is 5×10^{13} neutrons per cm² per second.

The atomic fuel is enriched uranium containing 5 per cent of U-235, the total charge being 550 kg. The heat transfer system consists of two circuits: the water of the first circuit, which circulates through the reactor, is under a pressure of 100 atm and

gives up its heat through a system of heat exchangers to the water of the second circuit, which, transforming into steam, drives a 5000-kw turbogenerator.

In this reactor the burnout of uranium 235 is 15 per cent. Owing to the low resonance capture the production of plutonium from uranium 238 is low and equals only 0.32. Fast neutron multiplication is practically absent. Thus, the reactor of the atomic station described works almost exclusively by burning uranium 235, the degree of enrichment decreasing during the operation of the reactor from 5 to 4.2 per cent.

The principal features of the reactor design were determined by the properties of the materials chosen. The reactor is encased in a hermetically sealed cylindrical steel jacket mounted on a concrete foundation. The jacket is filled with graphite brickwork, with proper clearance. To avoid burnout of the graphite the jacket is filled with helium (or nitrogen).

A total of 128 fuel channels pierce the central part of the graphite brickwork. Each fuel channel is a long graphite cylinder containing thin-walled steel tubes carrying the primary circuit water. The water passes through the upper head of the channel connected with the inlet and outlet headers, then runs down through the tubes and returns up, flowing over the surface of the uranium fuel elements.

The uranium fuel elements form a cylindrical active zone 150 cm in diameter and 170 cm high, enclosed by a graphite reflector.

To prevent rapid filling of the reactor with water in case of a rupture of channel tubes, which would lead to a rapid rise of reactivity, each channel is equipped with a cut-off device and a non-return valve to interrupt the flow of water from the header. A small water flow remains to remove the residual heat produced by radioactive decay of the fission products.

To compensate the excessive activity of the reactor, 18 boron carbide rods are installed in the reactor, 6 near its center and 12 at the extremities of the active zone. These rods can move in special water-cooled channels with a separate cooling system. There is a separate cooling system also for the graphite reflector.

The water of the primary circuit is heated in the reactor channels to a temperature of 260–270 C and passes to the steam generators from the header. There are eight such steam generators coupled in pairs in the shielded boxes. At peak power output, three pairs of steam generators are in operation and the fourth pair

is in reserve. Each steam generator consists of a water heater, a water evaporator and a steam-superheater. The water of the primary and secondary circuits in the steam generators moves according to the counter-current principle.

The water of the primary circuit leaves the steam generators at a temperature of about 190 C and enters the inlet pipe of the main circulation pumps, of which there are four. Three are in operation at peak power output, insuring water consumption of about 300 metric tons per hr; the fourth pump is in reserve.

There are special feed pumps for adding fresh water to the primary circuit. These pumps, with the aid of an automatic device, feed the water to seals of the main circulation pumps in such a way that the pressure of this water is always 0.5 atm above the water pressure in the pump. This insures a hydraulic seal which prevents the radioactive water in the main pumps from penetrating along the shaft.

It should be noted that as the water of the primary circuit moves away from the reactor, its activity gradually falls. The activity of the water coming out of the reactor is due to oxygen activity (0.2 curie per liter) with a short decay period (7 sec), whereas at the inlet of the reactor the activity of the water (2.10–5 curies per liter) is due basically to activity of impurities.

The water of the secondary circuit (the condensate) is fed by feed pumps to the heater of the steam generator; it then enters the evaporator, in which the level is maintained constant by a special automatic device; from the evaporator the steam enters the steam superheater. At peak power output of the station, the three operating groups of steam generators produce 40 metric tons of steam per hour at a pressure of 12.5 atm and at a temperature of 255–260 C.

Operating Experience

Since the atomic power station was started up on June 27, 1954, it has been tested in various operation conditions and has produced about 15 million kwhr of electric energy. This operation experience has made possible the following conclusions:

1. During operation of the station there has not been a single case of the fuel elements failing. This proves that the heat-transfer from the uranium elements to the water proceeds regularly without disturbances, and that the stainless steel selected for the tubes is a reliable material for work in the active zone of the reactor. De-

spite the fact that preliminary laboratory tests of the fuel elements made such results probable, a mass test, in the reactor of the station, of these most important parts might have produced undesirable contingencies. The durability of the fuel elements insured reliable operation of the electric power station as a whole.

2. The main circulation pumps, booster pumps, steam generators, volume compensators, worked without failure. This is due, to a considerable extent, to the fact that during installation of the high-pressure pipe lines, great attention was paid to the quality of welding. Samples of welded joints were studied to check absence of crystallite corrosion.

High purity of the water of the primary circuit is necessary for successful operation of the reactor. Bi-distillate, with which the primary circuit was filled, has a dry residue of less than 1 mg per liter. In the primary circuit, the content of dry residue is maintained at a level of 3 mg per liter by constant removal of the primary-circuit water and by adding up to 300 liters per hr of fresh bi-distillate. An increase in the dry residue is probably caused by washing away of copper gaskets and asbestos-graphite seals in the valves.

3. The equipment of the secondary circuit is of conventional type. But it was important to know how flexible the control of the whole scheme of the station was. At first there was a little uncertainty in the controlability of the whole reactor—steam generator—turbine setup, especially in transition operation conditions. A study of startup conditions established the fact that startup from a cold state, for example, after being shut down for repairs, may be carried out in 1.5 to 2 hours; the steam during this period is delivered to the startup condenser.

4. During operation of the station, a detailed study was made of the efficiency of the biological protection. It turned out that the gamma radiation intensity in the station, at nominal reactor power, was considerably less than the biological tolerance dose. When the apparatus is being unloaded the protective walls of the station completely safeguard work in the surrounding rooms. Protection of the rooms of the steam generators, the pumps and pipelines of the primary circuit likewise safeguard the service rooms from radiation.

During the time the station has

been operating there has not been a single case of over-irradiation of personnel. Each worker at the station has control dosimetric photo plate-holders and undergoes periodical medical examination. No harmful influence of the work of the station on the health of the workers has been observed.

Personals

Election of **E. P. Allis,** president of The Louis Allis Company since 1945, to the post of chairman of the board and promotion of **John W. Allis** from vice president to president, was announced by the board of directors of The Louis Allis Company.

Dr. B. D. Thomas, David C. Minton, Jr., and **John S. Crout,** associate directors of the Battelle Institute research organization, have been appointed vice presidents.

To speed development of "nuclear plumbing"—valves and fittings designed for use in nuclear power systems—Walworth Company is establishing a new research and development division at Boston. **Dr. Forest C. Monkman, Jr.,** of Boston has been named director of the program.

J. P. Wiseman, president of the Fluor Corporation of Canada, Ltd., for the past two and a half years, has been named to head a newly established products company of the Fluor Corp., Ltd., and **J. G. Bounds,** a member of the Fluor organization since 1939, was named to succeed Wiseman as president of the Canadian affiliate.

Otmar E. Teichmann, associate manager of the heat-power research department at Armour Research Foundation of Illinois Institute of Technology, has been named manager of the department to succeed **Wilson P. Green,** who resigned to accept a position in industry.

The promotion of **James E. Travis** to manager of the AEC's Hanford Operations Office, Richland, Washington, has been announced by **Kenneth E. Fields,** general manager of the Atomic Energy Commission.

Donald H. Larsen has been appointed fuel consultant, northern New York area, for the United Eastern Coal Sales Corp.

Robert S. Reaves has been made assistant to the vice president, director of engineering, tractor group, Allis-Chalmers Mfg. Co. succeeding **Harold W. Schudt,** now president of Canadian Allis-Chalmers, Ltd., an Allis-Chalmers subsidiary.

Carroll F. Hardy, after eighteen years in the employ of Appalachian Coals, Inc. (the last thirteen as chief engineer), has accepted a position with National Coal Association as director of sales engineering.

Dr. George A. Agoston, former senior research engineer with California Institute of Technology, has joined the physical sciences division of Stanford Research Institute as a senior physicist.

Frank L. Wilhelm has been elected assistant vice president of The Rust Engineering Co. with his headquarters in the company's home office in Pittsburgh where he will handle operational duties in connection with the company's activities.

Engineering and construction operations of The Fluor Corp. are being combined into one division under the direction of **M. A. Ellsworth,** vice president and general manager. Ellsworth had served as vice president in charge of the sales division since November, 1952.

George N. Decker has been named first vice president of the Kellogg Division, American Brake Shoe Co.; **William H. Starbuck** has been named vice president of the Sintermet Division and **Fred L. Cogswell** has succeeded Mr. Starbuck as vice president of the Kellogg Division.

H. B. Lammers, Jr., will join the staff of Appalachian Coals, Inc., as chief engineer succeeding **Carroll F. Hardy.** Mr. Lammers has been chairman and director of engineering for the Coal Producers Committee for Smoke Abatement.

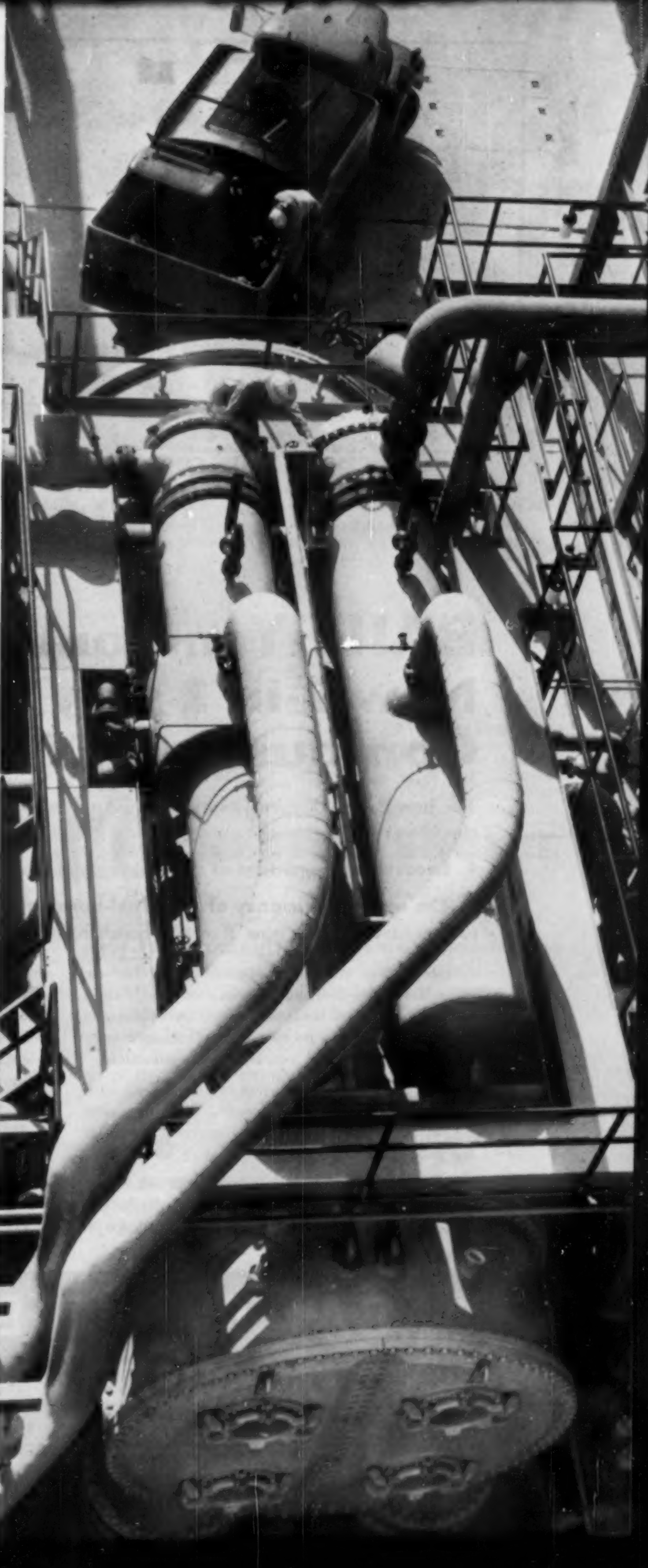
Le Roi Div. of the Westinghouse Air Brake Co. has appointed **H. J. Buttner** as manager of engineering.

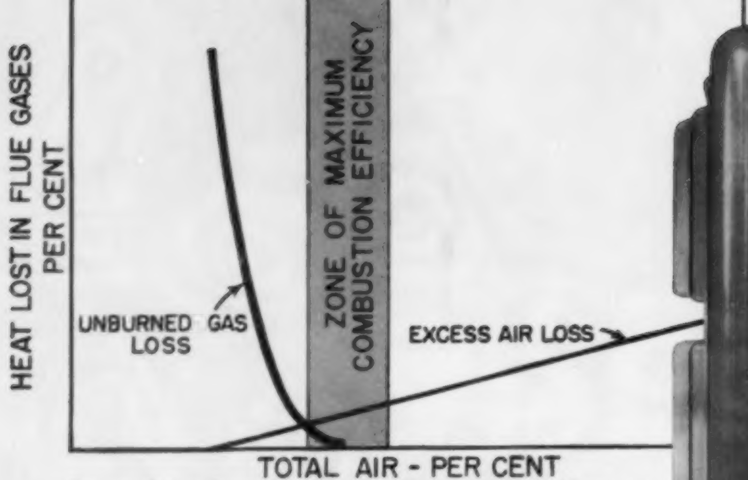
Obituaries

John C. Mahoney, superintendent of General Electric Company turbine installations for many years prior to his retirement in June, 1943, died in Schenectady this past August. Mr. Mahoney was widely known throughout the utility industry.

John H. Collier, 70, former president, and chairman of the board of Crane Co., died during the summer at his home in Fairfield, Connecticut, after a long illness.

Herbert J. French, 62, vice president of The International Nickel Co., Inc., died August 17, 1955 after an extended illness.





The new Bailey Oxygen-Combustibles Analyzer-Recorder (shown at right) provides a continuous two-in-one check of combustion efficiency by recording both oxygen and combustibles in flue gas. As shown by above chart, both measurements are needed to determine combustion efficiency.

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New 2 in 1 way to measure Combustion Efficiency

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Now, for the first time, you can check *both* with a *single* fast acting instrument, using the new Bailey Oxygen-Combustibles Analyzer-Recorder for industrial furnaces, kilns, heaters and boilers.

Fuel economy improves as excess air is reduced—until unburned fuel begins to show up in the flue gas. When this happens, combustion efficiency drops off sharply if there are further decreases in the air-fuel ratio. That's why combustion gases must be analyzed for *both* oxygen and combustibles to get a true indication of efficiency—and that is why Bailey coordinates both measurements on the same chart, to show when excess air may be reduced safely without danger of greater losses from unburned gases.

The Bailey Oxygen-Combustibles Analyzer is an approved combustion safeguard.

Ask your local Bailey engineer for suggestions on application. Equipment details in Product Specifications E65-1 and E12-5. P31-1

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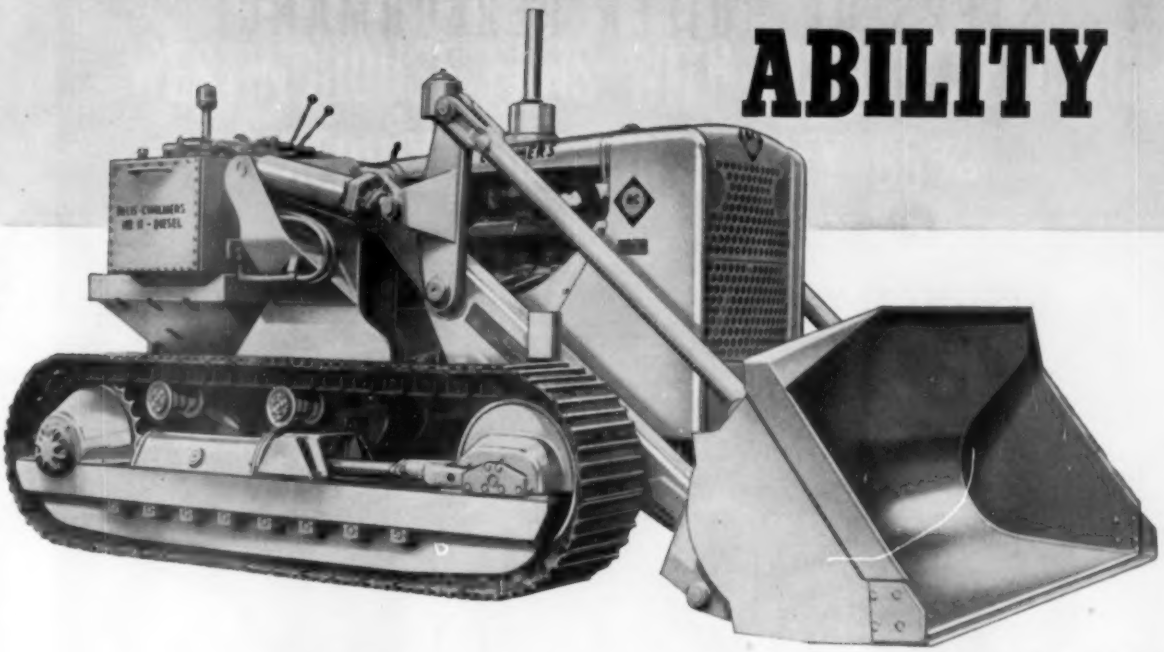

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NEW STABILITY—Extra-long track — with almost 9 ft of track on the ground — plus 8 truck wheels on each side and 32,000 lb of weight, give the HD-11G unusual stability, plenty of traction.

NEW-TYPE HYDRAULIC SYSTEM is simpler, safer, easier to maintain. Tank has fewer external fittings. Magnetic filters and suction-line screens protect entire system. Hydraulic pump offers fast action, long life.

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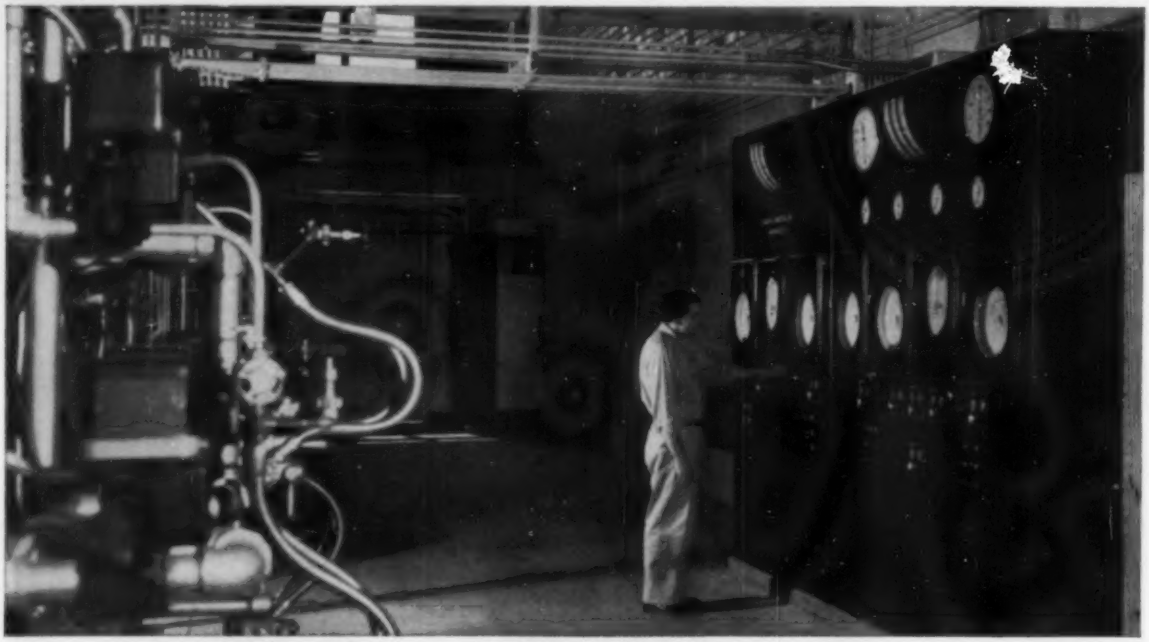

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(Continued on page 91)

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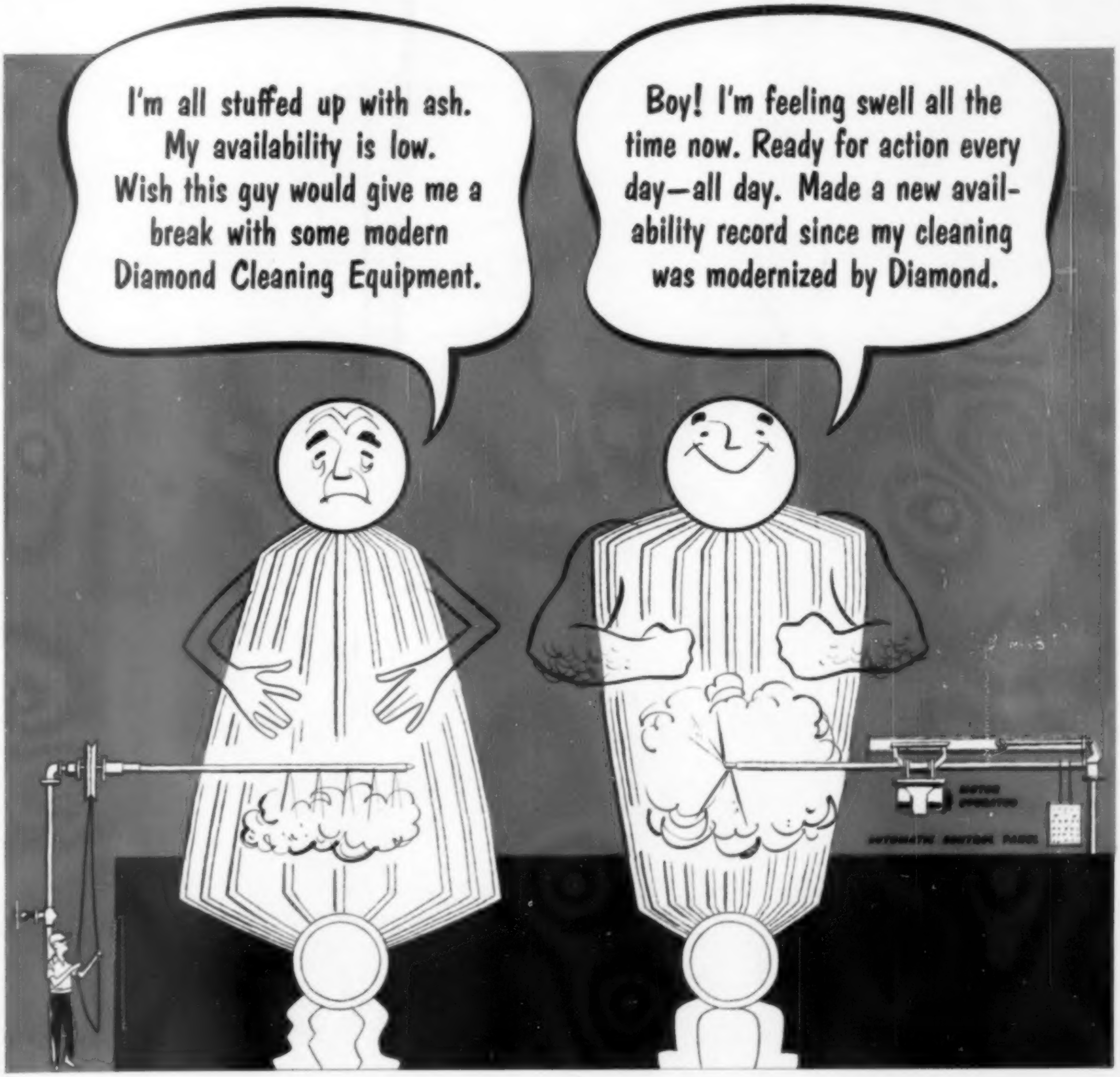

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